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Courtesy of Carrier Corporation, Syracuse, N. Y.

HEATING and VENTILATING

AIR CONDITIONING

A Home-Study Course and General Reference Work on the Principles, Design, Selection, and Application of Heating and Air-Conditioning Appliances and Systems for Residential, Commercial, Industrial Use.

Illustrated

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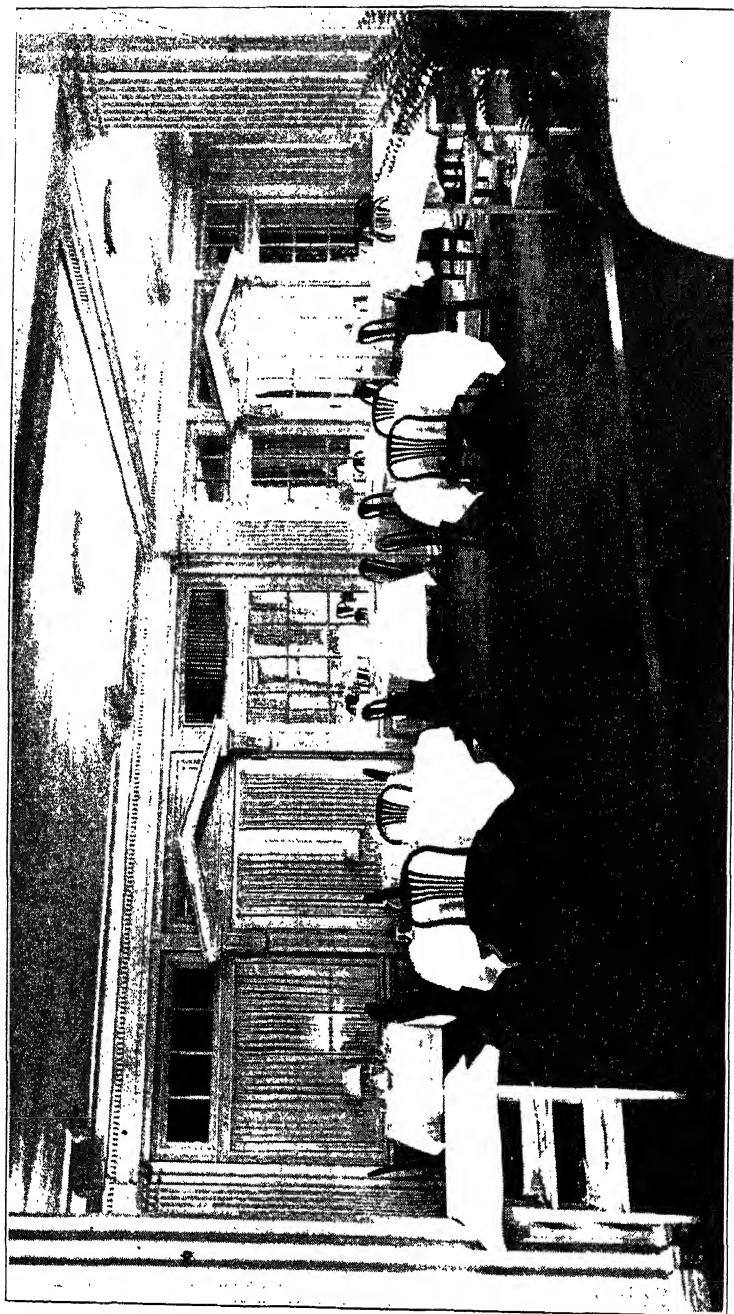American School. Co-author of *Logarithms*

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AUTHORITIES CONSULTED

THE EDITORS *have consulted standard technical litera-ture and expert engineers in the preparation of these volumes. They desire to express their indebtedness par-ticularly to the following authorities and organizations:*

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Secretary of the American Society of Heating and
Ventilating Engineers
✻

H. S. SHARP
The Henry Furnace and Foundry Company
✻

J. N. CRAWFORD
The Bryant Heater Company
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A. W. WILLIAMS
Secretary of The National Warm Air Heating and Ventilating
Association
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GEORGE R. METZGER
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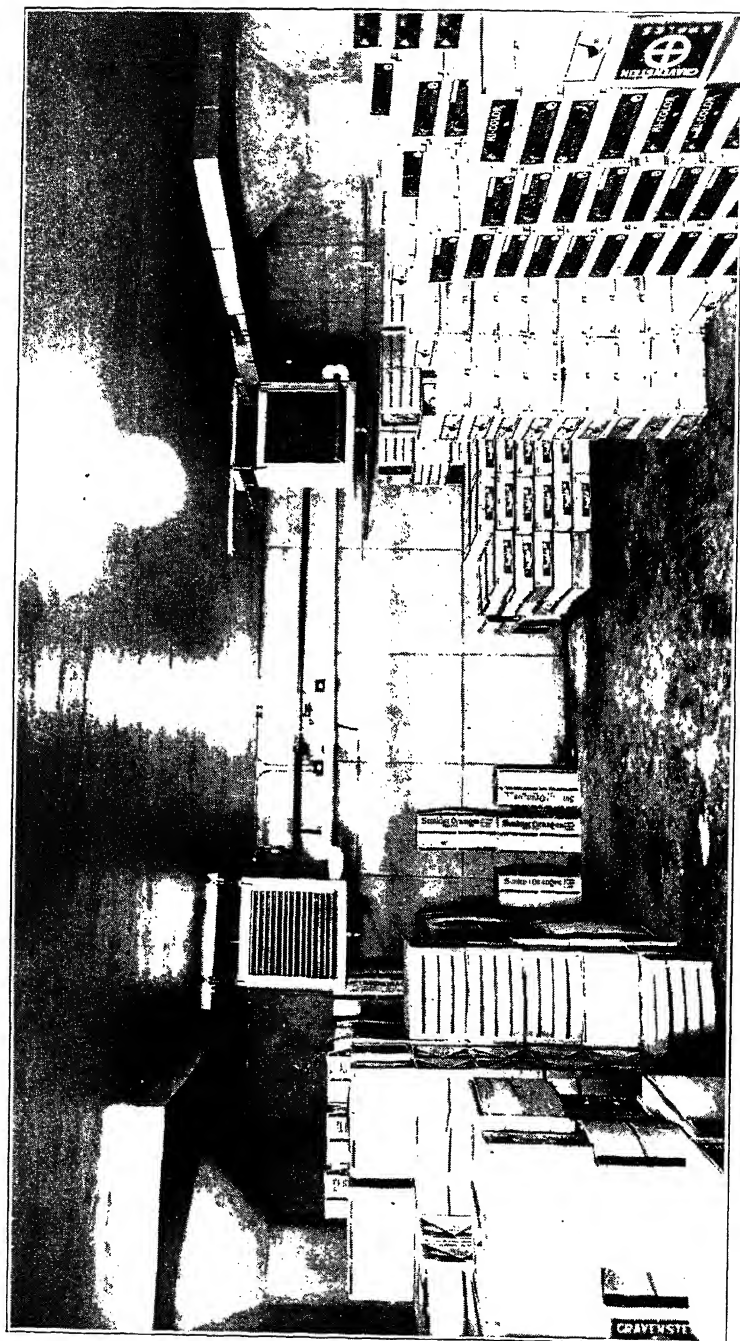The Carrier Engineering Corporation
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R. E. PEQUIGNOT
Electric Air Heater Company
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H. H. PEFFER
The American Society of Refrigerating Engineers
The Industrial Press

TRANE PRODUCT COOLERS IN THE STORAGE ROOMS OF THE FOX & GODDING COMPANY, CHICAGO

FOREWORD

★AIR CONDITIONING is not a new idea. As early as
1911 Willis H. Carrier had formulated some of the
principles and laws which are being used in present
day air-conditioning engineering. However, he prob-
ably did not dream of the possibilities in the field to
which he contributed.

★At first, air conditioning was developed only for
use in factories where the control of humidity and
where summer cooling permitted the continuation of
processes previously confined to the cool months of
the year. Gradually, the new applications of old
principles have brought the possibility of year-round
ideal manufacturing conditions in industrial plants.

★The success achieved in industrial air conditioning
suggested the possibilities in conditioning primarily
for comfort. Theatres, restaurants, and stores—
which always had suffered a hot-weather decline in
business—offered a fertile field. In theatres, the
cooled air increased business during the summer to a
point never before known. Restaurants and stores
also enjoyed the new summer prosperity. Thus air
conditioning for comfort was established. Public
demand for greater summer comfort, and the success
of air-conditioning engineers in achieving it, gave
impetus to the work of residential air conditioning.

★Few industries have enjoyed as rapid a development as air conditioning, and few industries have such potentialities. Air conditioning includes the treatment of air in one or a combination of several of the following ways: heating, cooling, humidifying, dehumidifying, ventilating, and cleaning. The air-conditioning industry is demanding trained men to carry on the work of improving and installing all types of systems, from those used in factories to those used in homes. The training necessary for this work is peculiar to the industry and highly technical.

★These volumes aim to meet the needs of men who, with the demand of the industry for specialized training in mind, look forward to joining those who are building the air-conditioning industry.

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†For professional standing of authors, see list of Authors and Collaborators at front of volume.
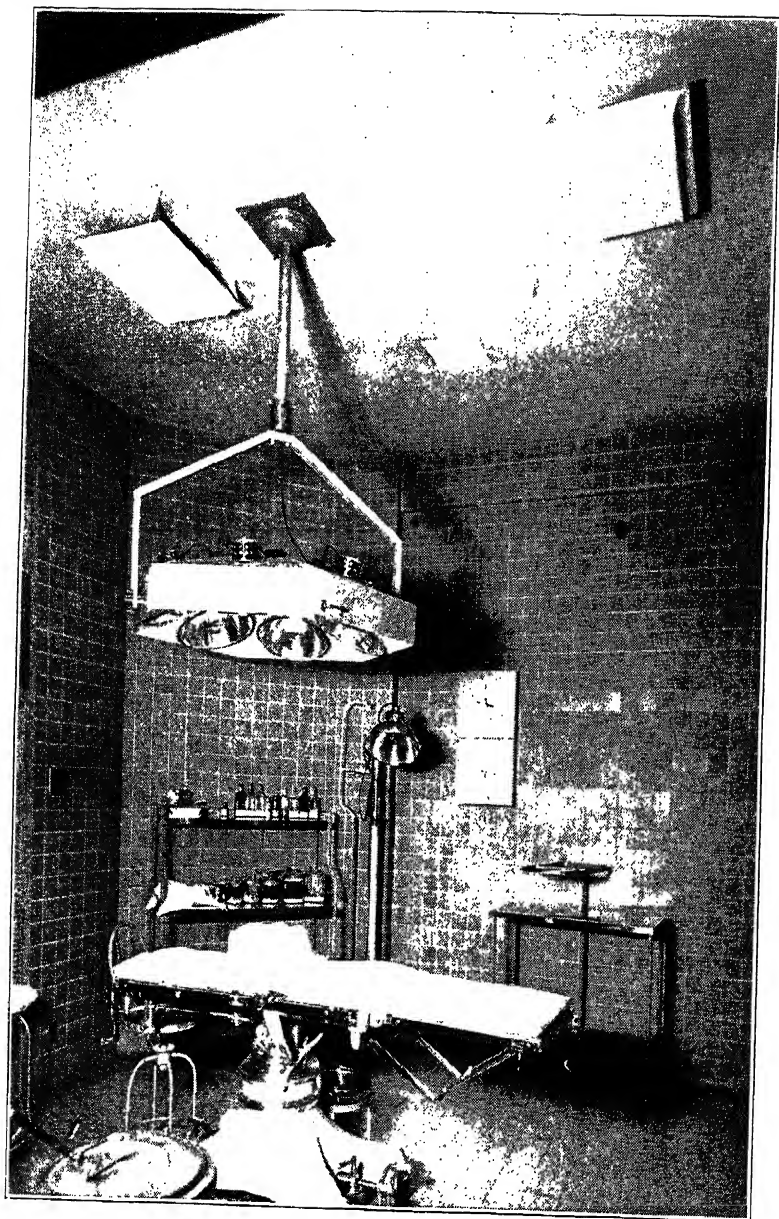
AIR-DISTRIBUTION DUCTS INSTALLED IN SPRINGFIELD CITY HOSPITAL,
SPRINGFIELD, OHIO

Courtesy of Carrier Corporation, Syracuse, N. Y.

HEATING AND VENTILATING

CHAPTER I

INTRODUCTION

A number of years ago Mark Twain, the Great American Humorist, in one of his comments on current events, said that "Everybody talks about the weather but nobody does anything about it." There is sufficient evidence to make one believe that behind this statement there was more truth than humor. There can be no doubt it was inspired by the hot, uncomfortable summer days along the Mississippi River in the Central States.

In the time of Mark Twain, summer meant days, weeks, and months of almost unbearable heat which had to be endured without hope of relief. Buildings were constructed of materials which offered little resistance to the intense heat of the sun. The structural bulks stored up heat during the day and released it at night. Thus the heat lasted through the nights, prevented proper relaxation and made impossible efficient living during the days.

Summer was accepted as a time of poor business. Shopping was made undesirable by the pent-up heat of stores. Theatres were even more undesirable due to the increased heat caused by the large number of people gathered in one enclosure. Restaurants were practically deserted because of hot-cooking odors, failing appetite, and the discomfort of being properly dressed. Railroad travel meant hours of extreme heat coupled with layers of cinders and dust. Motoring was little advanced. There was nothing left for people to do but to seek shade during the day and remain as quiet as possible at night. Frayed nerves were the natural result of these conditions of discomfort, too little sleep, and not much pleasant relaxation. It is no wonder that people talked about the weather.

During winter the discomforts caused by weather were of a different sort. They included hauling fuel, emptying ashes, attending to heating equipment, too much or too little heat, varying heat and, not least important, the ill effects of the dry heat supplied by

heating systems. This last item caused colds and discomfort even in enclosures where the air was 75°F., and a general bad effect on human bodies. People knew that furniture seemed to come apart during the winter, but the few who knew the cause did not know what to do about it.

Stores, theatres, restaurants, trains, and all public places became stuffy because in most cases they were closed as tightly as possible to keep out the cold. Heating systems caused damage to buildings and discomfort to people because of the circulation of coal dust, soot, gas, and common dust. The systems were usually so large they required most of the available basement space for supply and return pipes, fuel and ashes.

It seems that winter, like summer, gave ample cause for everybody to talk about the weather.

We have seen that the first part of Mark Twain's humorous remark was well founded. But how about the latter part of the quip—" . . . . nobody does anything about it"? The remark must have been wholly humorous because it is doubtful that Mark Twain, brilliant as he was, could have looked into the future to see what people *are* doing about it. The years since Mark Twain's day have seen people doing a great deal about the weather to make summer and winter discomforts a thing of the past.

To start with, buildings and residences are now being constructed with insulation against summer's heat and winter's cold. Many forms of insulation have been invented and manufactured which have the ability to lessen to a great extent the flow of heat. The insulation which can prevent the summer sun from pouring heat into buildings can, in the winter, prevent manufactured heat from escaping to the outdoors. Insulation in bulk or rigid form is built in as an integral part of roofs, ceilings, walls, and floors. In some cases it entirely supplants old materials and provides structural advantages in addition to insulating qualities. Continued progress is being made in insulation with ever growing benefits.

Summer is no longer the period of discomfort and poor business it used to be. Homes may now be perfectly ventilated and have cool, dehumidified air, which is free from impurities. Days and nights can now be delightfully cool and refreshing. Stores, theatres, restaurants, and trains enjoy these benefits with none of the old discomforts.

Business goes on in summer as well as in winter. People feel better than they used to feel, they rest better, relax in comfort and are able to go about their business with full efficiency.

To be perfectly comfortable, air must be both dehumidified and cooled. The marvel of treating air in this fashion is now almost commonplace. The early systems used in summer air conditioning aimed only at cooling the air from an outdoor temperature of 95°F., for example, to 70°F.—which was then considered a comfortable temperature. It was first tried in theatres, where it proved a boon to business. On the other hand, the shock of going from a temperature of 95°F. to a temperature of 70°F. proved too severe. This brought about a more careful removal of moisture (dehumidification) and less actual temperature difference. In this later approved form, the drier air, with less temperature difference, proved much more comfortable at about 83°F. (inside temperature) than the earlier temperature of 70°F. The drier air takes up perspiration rapidly and one's own body helps in the air conditioning for comfort. In some cases conditioned air is actually heated after being dehumidified. Cooling and dehumidifying allow all windows and doors to be kept closed at all times. This, combined with the filters which form a part of the conditioning apparatus, keeps enclosures clean and free from bacteria and pollen.

Winter air conditioning supplies heat, humidity, ventilation, and clean air. How different from the systems of a few years ago! Heat of today is supplied in even and constant amounts as needed. Rooms steadily retain whatever temperature is desired. All rooms heat easily and thoroughly. No longer need there be cold rooms or rooms which will not heat. Fuel handling can be entirely eliminated and ashes become a thing of the past. Humidity is supplied in ample quantities to provide health, comfort, and protection of furniture. No soot, dirt, or gas is present. All this can be accomplished automatically. All that is necessary is that a pilot light be ignited once in the early fall. Thermostats and other instruments keep steady temperatures and fixed humidity. They even turn down the heat if one plans to be away all day.

Apparatus for cooling, heating, dehumidifying, humidifying, cleaning, and ventilating can now all be combined in one enclosed air conditioner which is small and can be tucked away in some out-of-

the-way corner of the basement or utility room.   Pipes are now rectangular, smaller than the old furnace pipes and can be entirely out of sight.

In industry, in addition to its comfort and health-giving qualities, air conditioning has made possible the year-round processing of many previously seasonable operations.   Chocolate candy can now be made during 95°F. or 105°F. outside temperature.   Cigars can be manufactured anywhere with easy workability of tobacco because of controlled and steady humidity.   Almost all forms of industry have benefited from air conditioning.

The design and installation of air-conditioning systems require that the designer be well acquainted with many principles and laws of nature.   This book is, therefore, dedicated to explaining these necessary elements and showing what people are doing about the weather.

CHAPTER II

FUNDAMENTALS OF AIR CONDITIONING

Moisture Content of Air. Air at varying temperatures contains moisture (water) in varying amounts, depending on the temperature of outside air and upon mechanical regulation of inside (heated) air. The measurement of such moisture is found by a unit of measure called the *grain* and is measured in terms of grains per pound of air. Moisture in the air plays a prominent part in all heating and air-conditioning work. The proper functioning of the human body is partly conditioned by inside and outside temperatures, and also by the amount of moisture in the air. Too little moisture in heated air brings about discomfort in the winter; and too much moisture during the summer also brings about discomfort.

Saturated Air. For every possible temperature of natural air there is a corresponding weight or amount of water or moisture per pound of air. Thus air at any temperature has a definite amount of moisture it can hold or absorb. If, for any reason, the moisture content of the air is not up to the maximum amount at some given temperature, it is *dry* and seeks added moisture from surrounding objects. When it has attained its maximum moisture content, it is known as *saturated air*. It should be pointed out that saturated air at 70°F. contains a different number of grains per pound than does saturated air at 80°F., etc.

Humidity. Humidity is water vapor (moisture) mixed with dry air.

Absolute Humidity. Absolute humidity is the actual density of the water vapor in a mixture of air and water.

Superheated Water Vapor. If the temperature of an air-moisture mixture is higher than the saturation temperature, its vapor is called *superheated*.

Specific Humidity. In some cases it simplifies problems dealing with dry air-water vapor mixtures to express the weight (*specific humidity*) of the vapor in terms of the dry air.

Relative Humidity. Relative humidity, usually expressed in

per cent, is a ratio used to indicate the degree of saturation existing in a given space resulting from the water vapor present in that space. In other words, relative humidity is a ratio between the actual amount of moisture in a given volume of air and the amount of moisture that would be necessary to saturate that volume. (See Psychrometric Chart in back of book.)

Relative humidity will be dealt with to a great extent in all problems dealing with heating and air conditioning because of the moisture importance in any air used for human comfort. It is generally known that during the winter humidified air is necessary for healthful and comfortable conditions in any room or enclosure intended for human occupancy. Two examples will illustrate this point.

First, if inside air is not properly humidified in rooms during the winter months, it has a harmful effect on respiratory organs and makes them susceptible to the inroads of disease-carrying germs. Second, improperly humidified air has a great affinity for moisture and takes moisture from every available source. The slight, ever present moisture on the surface of human skin becomes just such a source of moisture. As the evaporating process is brought about by the dry air, the skin loses heat and a chilly sensation results, even within rooms where the temperature is 75°F. On the other hand, air supplied with the proper amount of moisture has no affinity for water and comfortable conditions can be had at 66° to 70°F.

During the summer when the temperature mounts to 90°F., or more, it often happens that relative humidity is also high. This brings about uncomfortable and unhealthy conditions, too. The human body functions so as to keep the body temperature below a certain point. There are three ways in which this is done but the method whereby sweating is caused is the only one of interest here. As the temperature of the body rises during periods of exercise or abnormally warm periods, the sweat glands cause the skin to be covered with moisture, which evaporates and causes cooling. This process depends on the air for its success in that the air causes the necessary evaporation. However, if the air is already high in relative humidity (near to saturation point), it has little or no ability to absorb the moisture on the skin. Thus very little evaporation takes place with the result that the body does not lose very much heat. In such conditions one feels the heat and is generally uncomfortable. This con-

dition is also a menace to health because the body heat should not be above a certain temperature. Therefore the humidity control of all rooms or other enclosures becomes as important in the summer as in the winter.

Control of Relative Humidity. The American Society of Heating and Ventilating Engineers recommends four general ways of controlling relative humidity as follows:

(a) If constant room temperature is to be maintained:

1. To maintain a constant relative humidity, the dew point must be kept constant.

2. To increase the relative humidity, the dew point must be raised.

3. To decrease the relative humidity, the dew point must be lowered.

(b) If constant dew point is to be maintained:

1. To maintain a constant relative humidity, the room temperature must remain constant.

2. To increase the relative humidity, the room temperature must be lowered.

3. To decrease the relative humidity, the room temperature must be raised.

(c) With varying dew-point temperatures:

1. To maintain a constant relative humidity, the room temperature must vary directly and in almost equal amount with the dew point.

2. To increase the relative humidity, the difference between room temperature and dew point must be decreased.

3. To decrease the relative humidity, the difference between room temperature and dew point must be increased.

(d) With varying room temperatures:

1. To maintain a constant relative humidity, the dew point must vary directly and in almost equal amount with the room temperature.

2. To increase the relative humidity, the difference between dew point and room temperature must be decreased.

3. To decrease the relative humidity, the difference between dew point and room temperature must be increased.

Dew Point. Dew point is the temperature of air at which any reduction in temperature will cause some of the vapor to be condensed or "squeezed out." This principle is made use of to a great extent in dehumidifying air during the summer.

Pressure Measurement. Force as applied to units of area is called pressure and is expressed in pounds per square inch, pounds per foot, inches of mercury, inches of water, atmospheres, etc. Steam gauges, for example, read pressures below atmospheric pressure or above it. A gauge recording pressure below atmospheric is called a vacuum gauge. Then the absolute pressure is the sum of the gauge and atmospheric pressures. For example, atmospheric pressure is 14.7. This is often assumed as 15 for ease in calculation. Then absolute pressure is gauge pressure plus 15. Most steam tables give readings in terms of absolute pressures.

Partial pressure refers to the pressure of any one of the many items composing air. The total pressure of air, as given above, is 15. Thus each gas such as oxygen, nitrogen, etc., has what is known as *partial pressures* because each is only a fractional part of the whole atmospheric pressure.

Temperature Measurement. Heat-energy is understood to lie in the rapid irregular vibrations of molecules of which all matter is composed. The molecules actually move rapidly from one body to another; or, in other words, heat may flow from one body to another. The conditions that determine which direction the flow will take constitute *temperature measurement.* Temperature, then, has nothing to do with quantity of heat. If a spoonful of water is dipped from a full pail, it is clear that the quantity of heat in each of the two measures of water differs, yet there is no transfer of heat from one to the other.

Equality of temperature may be estimated quite accurately simply by touching two bodies with the hand, provided they are of a similar nature and neither very hot nor very cold. The power which enables us to do this is called the temperature sense. It does not help us much, however, if the bodies are of a very different nature, nor does it tell us whether they are actually hot or cold. The sensation received really depends on the rate at which heat is communicated to (or taken from) the hand, and this depends on the temperature of the hand as well as on the nature of the material tested.

A simple experiment will illustrate this. Place the right hand in ice-water and the left hand in hot water; after a minute withdraw them and place them simultaneously in water just drawn from the faucet. It will seem warm to the right hand and cold to the left

hand, because in the first case heat passes from the water to the hand, and in the second case from the hand to the water. Again, a stone in winter feels colder to the hand than a piece of fur or woolen cloth. The stone conducts the heat away from the hand faster than the fur does, and thus gives the sensation of a lower temperature.

Thermometers. Instruments for the measurement of temperature are called thermometers. In designing a thermometer we may use any substance one of whose properties varies continuously with the temperature. Among the properties most convenient for use are:

1. Expansion; used for ordinary temperatures.
2. Change of electrical resistance; used for very low temperatures.
3. Thermo-electric effects; used for very high temperatures.

Besides these, many other properties of substances that depend on temperature are useful in special cases. For example, when a piece of polished steel is heated, its surface changes color, each color corresponding to a certain definite temperature. The process of tempering consists in heating the previously hardened articles until they assume the proper temperature, as shown by their color, and then plunging them again into cold water or oil. In this way each piece is made to indicate its own temperature without possibility of mistake.

Liquid Thermometer. In the most common form of thermometer, temperature is measured by the expansion of mercury in glass. On the end of a glass tube of very fine bore, a bulb is blown, and the bulb and part of the tube are filled with mercury, Fig. 1. The whole is then heated until the mercury completely fills the tube, after which it is sealed and allowed to cool. The space in the tube above the mercury is thus entirely freed from air. Changes in temperature cause the mercury to expand or contract, and the liquid in the tube will rise or fall accordingly.

But the thermometer thus made is not yet ready for use. It must have its divisions properly spaced and in the right places on the tube. All thermometers for accurate work should have their scales engraved on the tube itself, and not on a plate attached to it. Before we can engrave the scale we must know at least two points on the stem that correspond to known temperatures. The two points commonly taken are known as the *freezing point* and the *boiling point.*

The freezing point can easily be found by putting the thermometer into a mixture of clean pounded ice (or snow) and water. The boiling point is found by immersing the whole thermometer in steam from boiling water. The freezing point is always the same under ordinary conditions, but the temperature of the boiling point rises or falls slightly as atmospheric pressure increases or decreases.
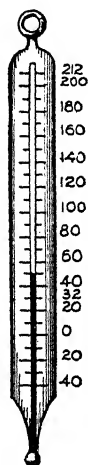
Fig. 1.

Dry-Bulb Thermometer

Fig. 2.

Taylor Sling Psychrometer. The advantage of this form of wet-and-dry-bulb hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as in whirling the bulbs they are subjected to perfect circulation. Consists of two accurately etched stem thermometers mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened

Two scales of temperature are in use. On the *Fahrenheit* scale, devised about 1714, the boiling point is marked 212°, and the freezing point 32°, there being thus 180 degrees between them. The *Centigrade* scale, which is more convenient for scientific work, has its boiling point marked 100° and its freezing point marked 0°.

We may convert Centigrade to Fahrenheit temperatures in the following way:

Since 100 Centigrade degrees cover the same temperature interval as 180 Fahrenheit degrees, one Centigrade degree is $\frac{180}{100}$ or $\frac{9}{5}$ as long as one Fahrenheit degree. Hence a temperature of m degrees

Centigrade is equal to $\frac{9}{5}\,m$ Fahrenheit degrees above the Centigrade zero. But this point is marked 32° on the Fahrenheit scale, consequently the total reading on the Fahrenheit thermometer will be

$$\frac{9}{5}m+32$$

The formula for changing a temperature C° Centigrade to its Fahrenheit equivalent F°, therefore, is

$$F°=\frac{9}{5}C°+32$$

and by transposing we obtain the corresponding formula,

$$C°=\frac{5}{9}(F°-32)$$

PRACTICE PROBLEMS

1. To what temperature F. does 58°C. correspond?   *Ans.* 136.4°F.
2. To what temperature C. does 149°F. correspond?   *Ans.* 65°C.
3. The difference between the temperatures of two bodies is 45°F. What is it in Centigrade degrees?   *Ans.* 25°C.
4. Lead melts at 327°C. What is its melting point on the Fahrenheit scale?   *Ans.* 620°F.

Temperatures below the zero point can be dealt with by calling them negative and using them with a minus sign.

Example. To what temperature F. does −20°C. correspond?

Solution. $F.°=\left(\dfrac{9}{5} \text{ of } -20\right)+32$

$= -36+32=4°.$ *Ans.*

5. To what temperature F. does −18°C. correspond?   *Ans.* −0.4°F.
6. To what temperature C. does −40°F. correspond?   *Ans.* −40°C.

Dry-Bulb Thermometer. A dry-bulb thermometer is one that is used to measure the temperature of a room or out of doors. In other words, the dry-bulb thermometer is used to determine the temperature of air.

Wet-Bulb Thermometer. A wet-bulb thermometer has its bulb covered with cloth which must be wet before the temperature is taken. The wet-bulb thermometer, Fig. 2, will give a lower reading for the same air than a dry-bulb thermometer in accordance with, or ratio to, the amount of evaporation from the cloth. The temperature

reading of a wet-bulb thermometer below that of a dry-bulb thermometer is called *depression* and measures the amount of moisture in the air. See Fig. 2.

Knowing both dry- and wet-bulb temperatures, the relative humidity can be quickly determined by use of the Psychrometric Chart.

*Psychrometric Chart. The Bulkeley Chart will be found attached to the inside back cover of the text on "Heating and Ventilating."

A·B·C·D·E = Directrix Lines
D.B.L. = Dry Bulb Line
D.P.L. = Dew Point Line
G.P.Lа. = Grains Moisture per Lb.Dry Air Saturated
T.H. = Total Heat per Lb. Dry Air Saturated
V.P. = Vapor Pressure in Mm. Mercury
G.P.C.F.S. = Grains Moisture per Cu. Ft. Saturated Air

R.H.L. = Relative Humidity Line
W.B.L. = Wet Bulb Line
S.L. = Saturation Line
W.P.C.F.S. = Weight per Cu.Ft.in Lbs. Saturated
R.D.S. = Relative Density per Cu.Ft.Saturated
W.P.C.F.D. = Relative Density per Cu.Ft.Dry
R.D.D. = Relative Density per Cu.Ft. Dry

$$\text{G.P.C.F.} \times \frac{\text{Abs.Temp.at D.P.}}{\text{Abs.Temp.at D.B.}} = \text{G.P.C.F. at Partial Saturation}$$

$$\text{W.P.C.F.} \times \frac{\text{Abs.Temp.at D.P.}}{\text{Abs.Temp.at D.B.}} = \text{W.P.C.F. at Partial Saturation}$$

$$\text{R.D.} \times \frac{\text{Abs.Temp.at D.P.}}{\text{Abs.Temp. at D.B.}} = \text{R.D. at Partial Saturation}$$
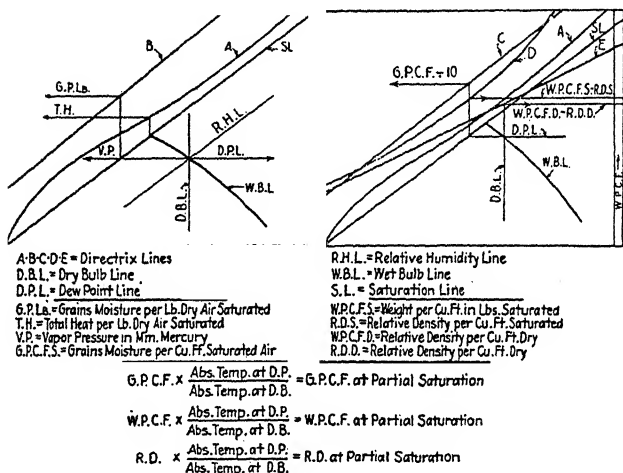
Fig. 3. Diagrams Showing Procedure to Follow in Using the Bulkeley Chart

Fig. 3 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

A is the total heat in B.t.u. contained in the mixture above 0°F., and is to be referred to the column of figures at the left side of the chart. Heat of the liquid is not included.

B is the grains of moisture of water vapor contained in each pound of the saturated mixture and is to be referred to the figures at the left side of the chart.

C is the grains of moisture of water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart which are to be divided by 10.

D is the weight in decimal fractions of a pound, of one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170° and 180°F. The relative density of the mixture is read in a similar manner from

the same curve by the column of figures between the vertical dry-bulb temperature lines 180° and 190°F.

E is similar to D but is for dry air, devoid of all moisture or water vapor. For convenience, the approximate absolute temperature of 500°F. is given at 40°F. on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.

ILLUSTRATIONS OF THE USE OF CHART

1. *Relative Humidity:* At the intersection of the 78°F. wet-bulb line and the 95°F. dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 46 per cent.

2. *Dew Point:* At the intersection of the 78°F. wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70.9°F.

3. *Vapor Pressure:* At the intersection of the 78°F. wet-bulb line and the 95°F. dry-bulb line, pass in a horizontal direction to the left of the chart and on the logarithmic scale read the vapor pressure as 19.4 millimeters of mercury. (Divide by 25.4 for inches.)

4. *Total Heat Above 0°F. in Mixture per Pound of Dry Air Saturated with Moisture:* From where the wet-bulb line joins the saturation line, pass in a vertical direction on the 78°F. dry-bulb line to its intersection with curve A and on the logarithmic scale at the left of the chart read 40.6 B.t.u. per pound of mixture. The use of this curve to obtain the total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of B.t.u. required to heat the mixture and humidify it, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

5. *Grains of Moisture per Pound of Mixture:* From 70.9°F. dew-point temperature on the saturation line, pass vertically to the intersection with curve B and on the logarithmic scale at the left read 114 grains of moisture per pound.

6. *Grains of Moisture per Cubic Foot of Mixture, Partially Saturated:* From 70.9°F. dew-point temperature on the saturation line proceed in a vertical direction to curve C, and on the logarithmic scale to the left read 83.3 which, divided by 10, gives 8.33 grains. A temperature of 70.9°F. is equal to an absolute temperature of 530.9, and 95°F. equals 555, absolute temperature. Therefore, $\dfrac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture.

7. *Grains of Moisture per Cubic Foot of Dry Air, Saturated:* Starting at the saturation line at the desired temperature, pass in a vertical direction to curve C and on the logarithmic scale at the left, read a number which, divided by 10, will give the answer.

8. *Weight per Cubic Foot of Dry Air and Relative Density:* From the point where, for example, the 70°F. vertical dry-bulb line intersects curve E, pass to right side and read 0.075 lb.; if cubic feet per pound are desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

9. *Weight per Cubic Foot of Saturated Air and Relative Density:* From the point where, for example, the 70°F. vertical line intersects the curve D, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70°F.

10. *Weight per Cubic Foot and Relative Density of Partially Saturated Air:*
Air at 50°F. and a wet-bulb temperature of 46°F. is to be heated to 130°F. The
wet- and dry-bulb lines intersect at a dew-point temperature of 42°F. Pass to
the left where this dew-point line intersects the saturation line and then pass
in a vertical direction to where the 42°F. dry-bulb line intersects with curve *D*.
Then pass directly to the right and read the weight per cubic foot of saturated
air at 42°F. as 0.07844 and the relative density as 1.046. The absolute tempera-
ture at 42°F. is 502, and at 130°F. is 590. Therefore, $\dfrac{502}{590} = 0.851$. The weight of
1 cu. ft. of air at 50°F. dry-bulb and 46°F. wet-bulb when heated to 130°F. is
$0.07844 \times 0.851 = 0.06675$, and the relative density is $1.046 \times 0.851 = 0.89$.

Properties of Dry and Saturated Air. Tables 1 and 2 give the
properties of dry and saturated air. The values in the tables may be
used at sea level and are commonly used for most other localities
except regions reading into high altitudes.

Effect of Altitude on Volume and Weight. The barometric
pressure varies according to different climatic conditions. Altitude
affects the barometer and the volume and weight of air. Near the sea
level an increase in altitude decreases the barometric pressure about
1 inch of mercury for each 900 feet of elevation. At altitudes of 4,000
to 6,000 feet this decrease in pressure is about 1 inch for each 1000
feet of elevation. Also, the barometer at a depth of 5,000 feet below
sea level should read as much *more* than 29.92 inches as it reads *less*
than 29.92 for 5,000 feet altitude.*

Vaporization. Since heat is the rapid, irregular, vibratory
motion of the molecules, it follows that if we add heat to a body we
increase this motion. At a certain stage the vibration is so vigorous
that the molecules (if the body is a solid) can no longer hold fast to
one another, and the solid literally falls to pieces, that is, it melts. By
applying more heat to the liquid and still further raising its tempera-
ture, we may finally reach a point at which some of the molecules are
moving so violently as to escape into the air, altogether free from one
another's influence. We then have a vapor, and the change into this
aëriform condition is called vaporization.

If vaporization takes place slowly, and only at the surface of a
liquid, it is called evaporation. Evaporation will be hastened by any-
thing that facilitates the escape of molecules from the liquid surface,
as by increasing the temperature of the liquid, lowering the pressure
on it, or causing a breeze to play over the surface.

*This data is from "Fan Engineering" by Buffalo Forge Co. A complete discussion of the
effect of altitude on volume and weight is given in "Fan Engineering."

The fact that heat is due to molecular motion explains why evaporation is a cooling process. Naturally those molecules will escape first whose motion is most violent, that is, whose temperature is highest. The more sluggish (and therefore colder) molecules stay behind. Thus, as the liquid evaporates, the departing molecules take with them more than their proportionate share of heat, and the remaining liquid grows colder.

Table 1. Properties of Dry Air
Barometric Pressure 29.921 Inches

Temp. Deg. F.	Wt. per Cu. Ft. Lbs.	Per Cent. of Vol. at 70°F.	B.t.u. Absorbed by 1 Cu. Ft. Dry Air per Deg. F.	Cu. Ft. Dry Air Warmed 1 Deg. per B.t.u.	Temp. Deg. F.	Wt. per Cu. Ft. Lbs.	Per Cent. of Vol. at 70°F.	B.t.u. Absorbed by 1 Cu. Ft. Dry Air per Deg. F.	Cu. Ft. Dry Air Warmed 1 Deg. per B.t.u.
0	.08636	.8680	.02080	48.08	130	.06732	1.1133	.01631	61.32
5	.08544	.8772	.02060	48.55	135	.06675	1.1230	.01618	61.81
10	.08453	.8867	.02039	49.05	140	.06620	1.1320	.01605	62.31
15	.08363	.8962	.02018	49.56	145	.06565	1.1417	.01592	62.82
20	.08276	.9057	.01998	50.05	150	.06510	1.1512	.01578	63.37
25	.08190	.9152	.01977	50.58	160	.06406	1.1700	.01554	64.35
30	.08107	.9246	.01957	51.10	170	.06304	1.1890	.01530	65.36
35	.08025	.9340	.01938	51.60	180	.06205	1.2080	.01506	66.40
40	.07945	.9434	.01919	52.11	190	.06110	1.2270	.01484	67.40
45	.07866	.9530	.01900	52.64	200	.06018	1.2455	.01462	68.41
50	.07788	.9624	.01881	53.17	220	.05840	1.2833	.01419	70.48
55	.07713	.9718	.01863	53.68	240	.05673	1.3212	.01380	72.46
60	.07640	.9811	.01846	54.18	260	.05516	1.3590	.01343	74.46
65	.07567	.9905	.01829	54.68	280	.05367	1.3967	.01308	76.46
70	.07495	1.0000	.01812	55.19	300	.05225	1.4345	.01274	78.50
75	.07424	1.0095	.01795	55.72	350	.04903	1.5288	.01197	83.55
80	.07356	1.0190	.01779	56.21	400	.04618	1.6230	.01130	88.50
85	.07289	1.0283	.01763	56.72	450	.04364	1.7177	.01070	93.46
90	.07222	1.0380	.01747	57.25	500	.04138	1.8113	.01018	98.24
95	.07157	1.0472	.01732	57.74	550	.03932	1.9060	.00967	103.42
100	.07093	1.0570	.01716	58.28	600	.03746	2.0010	.00923	108.35
105	.07030	1.0660	.01702	58.76	700	.03423	2.1900	.00847	118.07
110	.06968	1.0756	.01687	59.28	800	.03151	2.3785	.00782	127.88
115	.06908	1.0850	.01673	59.78	900	.02920	2.5670	.00728	137.37
120	.06848	1.0945	.01659	60.28	1000	.02720	2.7560	.00680	147.07
125	.06790	1.1040	.01645	60.79	1200	.02392	3.1335	.00603	165.83

Cooling by evaporation may be illustrated by a simple experiment. Drop about a teaspoonful of water on a table or smooth board, and set a small tin dish on the water. Pour three or four tablespoonfuls of ether into the dish, and blow upon it with a pair of bellows. After two or three minutes of vigorous blowing, the dish will be found frozen fast to the board. (**Caution.**—*Keep ether away from lights.* Ether vapor is highly inflammable.)

Table 2. Properties of Saturated Air

Weights of Air, Vapor of Water, and Saturated Mixture of Air and Vapor at Different Temperatures and at 29.921 Inches of Mercury

Weight in a Cu. Ft. of Mixture

Temp. Deg. F.	Wt. of the Dry Air Lbs.	Wt. of the Vapor Lbs.	Total Wt. of the Mixture Lbs.	B.t.u. Absorbed by 1 Cu. Ft. Sat. air per Deg. F.	Cubic Ft. Sat. Air Warmed 1 Deg. per B.t.u.	Specific Heat B.t.u. per Lb. of Mixture
0	.08625	.000068	.08632	.02083	48.02	.2413
5	.08528	.000087	.08537	.02060	48.54	.2414
10	.08433	.000110	.08444	.02039	49.05	.2415
15	.08339	.000139	.08353	.02018	49.56	.2416
20	.08246	.000176	.08264	.01998	50.07	.2418
25	.08154	.000222	.08176	.01978	50.57	.2419
30	.08062	.000277	.08090	.01958	51.07	.2420
35	.07970	.000339	.08004	.01939	51.58	.2423
40	.07878	.000409	.07919	.01921	52.06	.2426
45	.07786	.000491	.07835	.01903	52.55	.2429
50	.07694	.000587	.07753	.01885	53.05	.2431
55	.07601	.000699	.07671	.01868	53.54	.2435
60	.07506	.000828	.07589	.01851	54.02	.2439
65	.07409	.000978	.07507	.01835	54.50	.2444
70	.07310	.001151	.07425	.01819	54.97	.2450
75	.07208	.001350	.07343	.01804	55.44	.2457
80	.07103	.001578	.07261	.01790	55.87	.2465
85	.06993	.001838	.07177	.01775	56.34	.2473
90	.06879	.002134	.07092	.01762	56.76	.2485
95	.06760	.002470	.07007	.01749	57.18	.2496
100	.06635	.002850	.06920	.01736	57.59	.2509
105	.06503	.003279	.06831	.01725	57.98	.2525
110	.06364	.003762	.06740	.01714	58.35	.2543
115	.06217	.004305	.06648	.01704	58.69	.2563
120	.06060	.004914	.06551	.01695	59.00	.2587
125	.05893	.005594	.06452	.01686	59.31	.2613
130	.05715	.006351	.06350	.01679	59.56	.2644
135	.05524	.007191	.06243	.01673	59.78	.2680
140	.05319	.008120	.06131	.01668	59.96	.2721
145	.05090	.009153	.06014	.01664	60.11	.2767
150	.04864	.010295	.05894	.01662	60.17	.2820
155	.04612	.011553	.05767	.01661	60.21	.2880
160	.04340	.012936	.05634	.01662	60.17	.2950
165	.04048	.014451	.05493	.01663	60.14	.3028
170	.03734	.016108	.05345	.01668	59.96	.3121
175	.03397	.017919	.05189	.01675	59.71	.3228
180	.03035	.019896	.05025	.01684	59.38	.3351
185	.02646	.022053	.04851	.01695	59.00	.3495
190	.02228	.024400	.04668	.01710	58.49	.3663
195	.01780	.026950	.04475	.01730	57.81	.3866
200	.01300	.029715	.04272	.01749	57.18	.4094
205	.00783	.032707	.04054	.01767	56.60	.4359
210	.00230	.035938	.03824	.01802	55.50	.4712
212	.00000	.037307	.03731	.01815	55.10	.4865

When water is heated over a flame, the air (or any other gas) present is first driven off in tiny bubbles which rise to the surface and escape without noise. When the water nearest the flame is raised to the boiling point, bubbles of vapor are formed, which also rise through the water, but are condensed by the cooler layers before getting to the surface. This formation and condensation of steam bubbles produces

the sound known as singing or simmering. The "water hammer" in steam pipes is of a somewhat similar nature but on a larger scale. When the entire mass is heated to the boiling point, the steam bubbles rise to the surface and break, discharging their contents into the air with a characteristic noise. This stage is called *ebullition* or boiling.

Like air, steam is colorless, transparent, and invisible. What is commonly called "a cloud of steam" is really a cloud of fine water particles condensed from steam. Observe any steam jet, and notice that at the end of the pipe nothing whatever can be seen, the jet becoming visible only after it has gone far enough from the pipe to be cooled and condensed.

The increase of volume by vaporization is usually very great. For example, a cubic inch of water will make 1,661 cubic inches of steam at atmospheric pressure.

By increasing the pressure on the surface of a boiling liquid, we make it more difficult for the molecules to escape; they cannot escape unless given more motion, that is, unless they have a higher temperature than before. In other words, an increase of pressure raises the boiling point. Table 3 gives the boiling point of water under different pressures, as measured by a steam gauge:

Table 3. Boiling Point of Water

Gauge Pressure	Temperature, Fahrenheit
0 (atmosphere)	212°
50 lbs.	297.4
100 lbs.	337.6
150 lbs.	365.7
200 lbs.	387.8

Measurement of Heat. There are two units of measurement for determining quantities of heat. The British thermal unit (B.t.u.) is the amount of heat required to raise one pound of water from 59° to 60° Fahrenheit. The French unit, or calorie, is the amount of heat required to raise the temperature of one gram of water from 15° to 16° Centigrade. The former is much used in engineering calculations involving steam and fuels, and the latter in all other scientific work.

Latent Heat. If we put a block of ice into a vessel over a flame and insert a thermometer into the ice, we shall observe the thermometer rise to 0°C., at which point the ice begins to melt. The temperature of the ice and water then shows no further change until

all the ice has melted, though the heat is applied continuously. Only after the melting is complete will the temperature of the water begin to rise. It will then increase until 100°C. is reached, when ebullition begins, the temperature not rising above 100° until all the water has boiled away.

We thus see that in changing from ice to water and from water to steam there is absorbed a considerable quantity of heat which does not show on the thermometer. The quantities of heat absorbed in the processes of fusion and vaporization are called the *latent heat of fusion* and the *latent heat of vaporization* respectively.

The following example shows how the latent heat of fusion of ice may be measured. If we mix a gram of water at 80°C. with a gram at 0°, we get, as we should expect, two grams at 40°. But if we mix a gram of water at 80° with a gram of ice at 0°, we get two grams of water as before, but the temperature is 0° instead of 40°. The heat which in the first case raised the temperature of the water has in the second case been needed merely to melt the ice. The calculation of the latent heat is made in the following way:

One gram of water falling through 80 degrees of temperature will give out 1×80, or 80 calories. This quantity of heat is required to change one gram of ice at 0° into water at 0°. Therefore the latent heat of fusion of ice is 80; in other words, the heat which will just melt a quantity of ice will raise eighty times as much water one degree Centigrade.

By a somewhat similar method it is found that the latent heat of vaporization of water at atmospheric pressure is 536.5. That is, to evaporate one gram of water (already at the boiling point) will require as much heat as would raise the temperature of 536.5 grams one degree, or 5.365 grams from freezing to boiling (0° to 100°C.).

Expressed in terms of the Fahrenheit degree and the British thermal unit, the latent heats of fusion and vaporization are 144 and 970 respectively.

The large values of these quantities are of the greatest importance both in nature and in the arts. The great amount of heat necessary to melt the ice of winter makes the melting a slow process, and lessens the danger of destructive floods in the spring. In the autumn the water in freezing gives out again the heat absorbed in melting, and the transition to winter is thus rendered less abrupt.

Since a pound of steam in condensing will give out as much heat as 53.65 pounds of water cooling from 100°C. to 90°C., or from 90° to 80°, it follows that steam pipes for heating may be made smaller than water pipes for the same service. It also shows the value of steam as a carrier of heat; and in the arts advantage of this is taken in innumerable ways. The latent heat of water vapor in terms of B.t.u. per pound is calculated in the following formula:

$$LH = 1,092 - 0.56t$$

where LH = latent heat in B.t.u. per lb.

and t = temperature in °F.

Specific Heat. When equal quantities of different substances are raised equally in temperature, different amounts of heat are required; and in cooling through equal temperature intervals different substances give out different amounts of heat.

For example, if we mix a pound of water at 80°C. with a pound at 0°C., we get two pounds at 40°; but if we pour a pound of lead shot at 80° into a pound of water at 0°, the resulting temperature will be only 2.3°. A pound of lead, therefore, falling through 77.7 degrees of temperature, is able to raise a pound of water only 2.3 degrees. The fall of temperature of the hot body is nearly twice as great as in the first case, and the heat given out in the fall only about one-seventeenth as much. The heat capacity of the lead is therefore much less than that of the water.

If we know how much heat will raise the temperature of a given substance a certain amount, and how much is required to raise the temperature of an equal quantity of water by the same amount, then the ratio of these two quantities is called the specific heat of the substance. In other words, if we take the specific heat of water as our standard (as we practically do in defining the units of heat), the specific heat of a substance is expressed by the number of heat units required to raise the unit quantity of the substance one degree.

One of the simplest methods of determining specific heat is by mixing the substance with water. Suppose that 6 pounds of mercury at 100°C. are poured into 2 pounds of water at 0°C., and that the resulting temperature of the "mixture" is 9°. The specific heat S of the mercury can then be found as follows:

In falling from 100° to 9° the 6 pounds of mercury give out

$6\times(100-9)\times S$, or 546 S heat units.   These have gone to heat 2 pounds of water from 0° to 9°, which requires 2×9, or 18 heat units. Hence we may write

$$546S = 18$$

Therefore $$S = 0.033$$

Example. Half a pound of a metal at 212°F. is dropped into one pound of water at 68°F. The temperature of the mixture is then observed to be 76°F. What is the specific heat of the metal?   *Ans.* 0.117

Table 4.   Specific Heats

Hydrogen	3.4090	Steel	0.1181
Alcohol	0.6020	Wrought iron	.1124
Ammonia (gas)	.5084	Copper	.0949
Ice	.5040	Zinc	.0935
Air	.2375	Tin	.0562
Aluminum	.2122	Mercury	.0330
Glass	.1930 to .1980	Lead	.0314
Cast iron	.1298		

With the foregoing principles and the help of suitable tables, many problems can be solved. For example, let us find how many calories will be required to convert 10 grams of ice at —12°C. into steam at 100°C. (See Table 4.)

Solution. Required to raise the ice from —12° to 0°,

$$10\times12\times0.504 = 60.48 \text{ calories}$$

Required to melt the ice,

$$10\times80 = 800 \text{ calories}$$

Required to raise the water from 0° to 100°,

$$10\times100 = 1000 \text{ calories}$$

Required to vaporize the water,

$$10\times536.5 = 5,365 \text{ calories}$$

Total number of calories required,

$$60.48 + 800 + 1000 + 5,365 = 7,225.5 \text{ (nearly)}$$

PRACTICE PROBLEMS

1. What weight of water at 75°C. will just melt 15 pounds of ice at 0°?
                                        Ans. 16 pounds

2. One kilogram of water at 40°, 2 kilograms at 30°, 3 kilograms at 20°, and 4 kilograms at 10° are mixed. Find the temperature of the mixture.
                                        Ans. 20°

3. How many heat units will be required to melt 5 grams of ice at —20°C.? How many grams of water at 50°C. would do it?
                                        Ans. 450.4 heat units; 9.01 grams

If we wish to use Fahrenheit degrees and British thermal units in our calculations, it is necessary to remember that the numbers representing the heats

of fusion and of vaporization are different, but that the specific heat, which is a mere ratio, is the same in both systems.

For example, let us find how many B.t.u. are required to convert 12 lbs. of ice at 10°F. into steam at 212°F.

Solution. Required to raise the ice from 10°F to 32°F.,

$$12 \times 22 \times 0.504 = 133.056 \text{ B.t.u.}$$

Required to melt the ice,

$$12 \times 144 = 1,728 \text{ B.t.u.}$$

Required to raise the water from 32° to 212°,

$$12 \times 180 = 2,160 \text{ B.t.u.}$$

Required to vaporize the water,

$$12 \times 966 = 11,592 \text{ B.t.u.}$$

Total number of B.t.u. required,

$$133.056 + 1,728 + 2,160 + 11,592 = 15,613 \text{ (approx)}$$

4. How many B.t.u. are required to convert 10 lbs. of ice at 15°F. into steam at 212°F.? *Ans.* 12,985 B.t.u.

For ordinary purposes we may proceed as above; but as the specific heat and latent heat of water vary for different temperatures, we must, where great accuracy is necessary, employ a table of the properties of steam and water.

Properties of Steam. The relation between the external pressure and the boiling point of water is a perfectly definite one, but cannot be exactly expressed by any mathematical equation. In dealing with this and other properties of steam and water, it is therefore customary to refer to suitable tables where the values are given, as determined by experiment. Such tables are called steam tables, and are much used in engineering calculations. (See Table 5.)

In the following table are given (1) the pressure above absolute vacuum, (2) the corresponding temperature, (3) the amount of heat in B.t.u. required to raise a pound of water from 32°F. to the given temperature, (4) the amount of heat in B.t.u. required to vaporize a pound of water at the given temperature; (5) equals the sum of (3) and (4).

A steam gauge measures pressures above the atmospheric pressure; hence, when readings are taken from a steam gauge, the barometric pressure (averaging 14.7 lbs. per sq. in., or call it 15 lbs.) must be added to obtain the absolute pressure.

With a steam table we can extend considerably the range of problems. For example, let us find how many pounds of steam at 65 lbs. gauge pressure will be needed to raise the temperature of 60 pounds of water from 50°F. to 100°F.

Table 5. Properties of Saturated Steam[1]

1	2	3	4	5	6	7	8
		Specific volume cu. ft. per lb.			Total heat B.t.u. per lb.		
Absolute pressure, lb. per sq. in. p	Temp., deg. F. t	Sat. liquid V_f	Evap. V_{fg}	Sat. vapor V_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g
½" Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7
¾" Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0
1" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9
1½" Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6
2" Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8
2½" Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1
3" Hg	115.0S	0.01618	231.8	231.8	82.96	1027.9	1110.8
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0
4.0	152.99	0.01630	90.72	90.74	120.83	1005.9	1126.8
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5

[1]Abstracted from "Steam Tables and Mollier Diagram," by Prof. J. H. Keenan, 1930 edition, by permission of The American Society of Mechanical Engineers.

Solution. To raise one pound of water from 50° to 100° requires 50 B.t.u.; and for 60 pounds we need 50×60, or 3,000 B.t.u. At 65 lbs. gauge pressure (80 lbs. absolute) the total heat of one pound of steam is 1,177 B.t.u., and this amount would all be available if we cooled it down to 32°F. But since the cooling is not carried below 100°F., we cannot use 100−32, or 68 B.t.u., and the amount available is therefore 1,177−68=1,109 B.t.u.

The quantity of steam therefore needed is

$$\frac{3,000}{1,109}=2.705+ \text{ pounds}$$

Ans. 2.705+pounds

The small quantity of steam in this example well illustrates the great heating power of steam.

PRACTICE PROBLEMS

1. How many pounds of steam at 100 pounds absolute pressure will raise 250 pounds of water from 50°F. to 150°F.?   *Ans.* 23.5 pounds

2. How many pounds of steam at 35 pounds gauge pressure will just melt 1000 pounds of snow at 32°F.?   *Ans.* 123.3 pounds

Sensible Heat. Sensible heat is any heat that would be recorded on an ordinary dry-bulb thermometer. To find sensible heat the following formula is used.

$$H=S_H(t_2-t_1) \tag{1}$$

where

$H=$ sensible heat in B.t.u. per pound

$S_H=$ mean specific heat. (approx. 0.241)

$t_2-t_1=$ increase in dry-bulb temperature

Total Heat of Air. As already explained, air contains water vapor in amounts depending on temperature and degree of saturation. Therefore, to find total heat of air, the latent heat of the vapor is added to the sensible heat. The total heat may be calculated by the following formula which is in a simplified form.

$$h=0.24t+Wh_v+0.45W(t-t^1) \tag{2}$$

where

$h=$ total heat in 1 pound dry air plus vapor

$t=$ temperature water vapor and air

$W=$ weight of vapor in pounds

$h_v=$ heat of vaporization

$t^1=$ wet-bulb temperature

For air when fully saturated, the formula becomes

$$h = 0.24t^1 + Wxh_v \tag{3}$$

*RELATION OF DEW=POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew-point and the relative humidity and this is found most useful in air-conditioning work. This is, that for a fixed relative humidity there is substantially a constant difference between the dew-point and the dry-bulb temperature over a considerable temperature range. Table 6 giving the dry-bulb and dew-point temperatures, and dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

Table 6. Dry=Bulb and Dew=Point Temperatures for 50 Per Cent Relative Humidity

Dry-bulb temperature....................	65.0	70.0	75.0	80.0	85.0	90.0
Dew-point temperature...................	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew-point and dry-bulb temperature...........................	19.2	19.5	19.75	20.25	20.75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 degrees throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 degrees.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 degrees through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_1}{20}}} \tag{4}$$

where

$$t_1 = \text{dew-point temperature}$$

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity

is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills and tobacco factories, where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat to control the relative humidity in the room.

Table 300 of the Appendix gives, for different temperatures, the density of saturated vapor, d_t, the weight of saturated vapor mixed with 1 pound of dry air, W_t (at a relative humidity of 100 per cent and a barometric pressure, B, of 29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 pound of dry air (at a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The data from Table 300 of the Appendix may be used in solving the examples that follow.

NOTE: The Appendix will be found following the final chapter of "Heating and Ventilating."

[1]Example 1. *Humidifying and Heating.* Air is to be maintained at 70°F. with a relative humidity of 40 per cent (Φ[2]$=0.4$) when the outside air is at 0°F. and 70 per cent relative humidity ($\Phi=0.7$) and a barometric pressure, B, of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Table 300 in Appendix,

$$W_1 = 0.622 \left(\frac{0.7 \times 0.03773}{29.92 - 0.0264} \right) = 0.000548 \text{ lb. per pound of dry air}$$

$$W_2 = 0.622 \left(\frac{0.4 \times 0.7386}{29.92 - 0.295} \right) \quad 0.00618 \text{ lb. per pound of dry air}$$

The .03773 comes from Table 300 of the Appendix and is found in first column opposite +0°F. temperature. The .7386 is found opposite the 70°F. temperature. The .0264 is obtained by multiplying .03773 by .70 = .0264 and .7386 × .40 = .295.

The water vapor added per pound of dry air must be ($W_2 - W_1$) or 0.005632 lb. By inspection of Table 300 of the Appendix, $W_t = 0.00618$ at 44.5°F., so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 300 in Appendix is

$W_1 = 0.7 \times 0.0007852 = 0.00054964$ lb. per pound of dry air
$W_2 = 0.4 \times 0.01574 = 0.006296$ lb. per pound of dry air

[1]Use Formula $W = 0.622 \left(\dfrac{e}{B-e} \right)$ pounds.

where W = weight of vapor mixed with each pound of dry air
         e = actual partial pressure of vapor
         B = barometric pressure
[2]Means relative humidity.

The water vapor added per pound of dry air is approximately 0.00574636 lb. and the dew-point temperature is approximately 45°F. The degree of approximation is evident.

[1]**Example 2.** *Dehumidifying and Cooling.* Air with a dry-bulb temperature of 84°F., a wet-bulb of 70°F., or a relative humidity of 50 per cent ($\Phi = 0.5$), and a barometric pressure, B, of 29.92 in. of mercury is to be cooled to 54°F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Table 300 in Appendix,

$$W_1 = 0.622 \left(\frac{0.5 \times 1.1752}{29.92 - 0.5876} \right) = 0.01248 \text{ lb. per pound of dry air}$$

$$W_2 = 0.622 \left(\frac{0.42003}{29.92 - 0.42003} \right) = 0.00887 \text{ lb. per pound of dry air}$$

Calculated in the same manner as Example 1.

Since $W_1 = W_t$ when $t = 63.4°F.$, this is the dew-point temperature of the entering air. The weight of vapor condensed is $(W_1 - W_2)$ or 0.00361 lb. per pound of dry air.

An approximate result is

$W_1 = 0.5 \times 0.02543 = 0.012715$ lb. per pound of dry air.

$W_2 = 1 \times 0.008856 = 0.008856$ lb. per pound of dry air, since the exit air is saturated.

Since $W_1 = W_t$ at $t^2 = 64°F.$, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.003859 lb. per pound of dry air. The degree of approximation is again evident.

Adiabatic Saturation. When dry air is combined with water vapor, the temperature of the water is lowered below the temperature of the original dry air. After the combination has taken place, the air is the same temperature as the water. A heat transfer of this kind is called adiabatic saturation of the air.

The above principle of adiabatic saturation provides a method of ascertaining the amount of water vapor in the air by comparing wet- and dry-bulb temperatures. The following formula is used.

$$W = \frac{h_{va}W^1 - C_{pa}(t-t^1)}{+ C_{ps}(t-t^1)} \tag{5}$$

[1]Use Formula $W = 0.622 \left(\frac{e}{B-e} \right)$ pounds.

where W = weight of vapor mixed with each pound of dry air.
         e = actual partial pressure of vapor.
         B = barometric pressure.
[2]Means dry-bulb temperature.

where W = weight of water vapor mixed with 1 pound of dry air at temperature t.

h_{va} = latent heat of vaporization at temperature t^1

C_{pa} = specific heat of dry air at constant pressure

t = temperature of air

t^1 = temperature of adiabatic saturation or wet-bulb temperature

C_{ps} = specific heat of water vapor at constant pressure

Example 1. Assume wet- and dry-bulb temperatures of 69°F. and 81°F. Find (a) dew point, (b) percentage of saturation, (c) moisture content.

Solution. For this solution, use the Psychrometric Chart. (a) First, locate the dry-bulb temperature of 81°F. along the bottom of the chart. Follow a vertical line upward until it intersects the curved line for wet-bulb temperature of 69°F. Knowing that all horizontal lines are to the right-hand side of the saturated line (see description) it is a simple matter to trace the horizontal line nearest the above mentioned intersection to the right-hand side of chart and read the dew-point temperature of approximately 63°F.

(b) The point of intersection of the wet- and dry-bulb lines comes almost midway between the 50 and 60% relative humidity lines. This is called 54%.

(c) From the point where the dew-point horizontal temperature line intersects the saturated line, follow a line upward until it meets curve B. From this point of intersection, follow a horizontal line to the scale at the left-hand side of the chart which reads 88 grains per pound of dry air.

Example 2. Assume air at 60°F. containing 55 grains of moisture per pound of dry air. Find relative humidity.

Solution. Use Psychrometric Chart. Starting at the left-hand side of the chart, follow a line from 55 grains to the right until it intersects curve B and from that point follow a vertical line downward until it intersects the saturation line. From this point follow a horizontal line toward the right until it meets the vertical line representing a dry-bulb temperature of 60°F. This last point of intersection is approximately on the 70 per cent relative humidity line. The answer is therefore 70 per cent relative humidity.

Example 3. Assume outside air is 30°F. dry-bulb and 40 per cent relative humidity and that it is being supplied to a ventilation system. The air first passes through a tempering coil[1] where it is heated to 55°F. It next passes through an air washer.[2] The air leaving the washer is 100 per cent saturated. Next, the air passes through a reheating coil where the temperature is raised to 80°F. Find the relative humidity of the air leaving the reheating coil.

Solution. Refer to the Psychrometric Chart and follow a vertical line from 30°F. dry-bulb temperature to the intersection of this line and the 40 per cent relative humidity line and then horizontally to the saturation line. From the saturation line, follow a vertical line upward to curve B and from this point of intersection follow a horizontal line to the scale at the left-hand side of the chart. This reads 10 grains of moisture, approximately.

Note: While 10 is not the exact number of grains, it is used here for ease in calculation.
[1] See page 385.
[2] See page 309.

After leaving the tempering coil, the air is at a temperature of 55°F., and as it still contains the same amount of moisture, the process of finding the relative humidity at this stage is the same as explained in the previous example. It is found to be approximately 15.5 per cent and the wet-bulb temperature 39°F. Passing through the air washer the air is cooled to the wet-bulb temperature 39°F. and is 100 per cent saturated when leaving the washer. By following a vertical line, from the intersection of the 39°F. vertical line and the saturation line, where it intersects curve B, and then horizontally to the scale at the right-hand side of the chart, it is found that the air at this stage contains approximately 35 grains of moisture per pound. Finally, starting again at the intersection of the 39°F. and saturation lines, and following a horizontal line toward the right until it meets the 80°F. dry-bulb line it is found that the relative humidity of the air as it leaves the reheating coil is approximately 24 per cent.

PRACTICE PROBLEMS

1. The air outside is at a temperature of 45°F. and has a relative humidity of 81 per cent. When drawn into a building, it is heated to 70°F. What is its relative humidity at 70°F.?   *Ans.* 34 per cent

2. Assume air at 85°F. is 79 per cent saturated. When cooled to 65°F., what is its moisture content?   *Ans.* 92 grains per pound of dry air

3. A stream of air is sprayed with water, and moisture is thus added. The original condition is dry-bulb temperature of 85°F. and wet-bulb temperature of 70°F. The final dry-bulb temperature is 75°F. How much moisture was added per pound of air?   *Ans.* 16 grains

*GLOSSARY OF TERMS

To better understand the various terms that will be continually used and referred to throughout this book the following simple definitions will be helpful.

Dry-Bulb Temperature. This is the "first" temperature of the air and is measured by a thermometer of standard type.

Wet-Bulb Temperature. This is the "second" temperature of the air, and is a measure of the "degree" of moisture in the air. It is determined by a standard thermometer having its bulb wet by water and exposed to air circulation. It is a measure of the *total* heat in the air.

Effective Temperature. This is an experimentally determined temperature and is an arbitrary composite index of the "effect" on the human body of a combination of temperature, humidity, and air movement. (See page 56.)

Dew-Point Temperature. This is the temperature at which air becomes fully saturated with its present moisture content. This is always lower than the dry-bulb or wet-bulb temperatures, unless the air is saturated, in which case the dew-point, dry-bulb, and wet-bulb temperatures are identical.

Humidity is the amount of water vapor or moisture in the air.

Absolute Humidity. This is the actual amount of water vapor or moisture in the air; it is not affected by any change in temperature.

Relative Humidity. This expresses the percentage of "saturation" at a given temperature. For example, air that contains one-half of the amount of moisture it is capable of holding has a "relative" humidity of 50 per cent.

*Extracted from Bryant Air-Conditioning Manual.

Saturated Air. Air which has all the water vapor (humidity or moisture) it can hold. Such air has a "relative" humidity of 100 per cent.

Sensible Heat. The heat corresponding to the dry-bulb temperature. It is measured by a standard thermometer.

Latent Heat. In an air-conditioning sense, this is the heat liberated in changing a vapor to a liquid without any change in temperature.

Latent Heat of Vaporization. The heat absorbed in changing a liquid to a vapor—humidification.

Latent Heat of Condensation. The heat liberated in changing a vapor to a liquid—dehumidification.

Total Heat. The sum of both sensible and latent heats. The "wet-bulb temperature" is an exact indication of total heat.

Entering Temperature Differential. The difference in dry-bulb temperature between the room or inside air and the entering "conditioned" air.

Heat. A form of energy. Every substance contains heat and it flows from one substance to another, the rate of flow depending upon temperature differential.

British Thermal Unit (B.t.u.). The *quantity* of heat required to raise *one* pound of water *one* degree Fahrenheit.

Specific Heat. The *quantity* of heat required to raise the temperature of *one* pound of *any* substance *one* degree Fahrenheit (expressed in B.t.u.).

Grain of Moisture. The unit used as a measure for the weight of water vapor in air.

Vapor Pressure. The pressure exerted by the molecular activity of the gaseous molecules where the vapor is in a saturated condition.

Partial Pressure. This is pressure exerted by each gas in a mechanical mixture of gases, the total of which equals the observed pressure of the mixture.

Ton of Refrigeration. This represents the removal of heat at the rate of 12,000 B.t.u.'s per hour.

Adsorption. This designates the property of certain substances to condense water vapor without themselves being changed physically or chemically. Silica Gel is an example of such a substance.

Recirculation of Air. The recirculation of air is economical in simple heating systems or in winter and summer air-conditioning processes. Considering the fact that economy in installing and operating heating and air-conditioning systems is so important, this subject should always be considered in selecting equipment.

Heating Systems. When we think of the heating of an ordinary residence, we must presuppose a hot air furnace of a given size where *all air is recirculated.* The heated air is supplied to the rooms by the leaders, stack, or ducts and returned to the furnace by the cold air ducts. The leaders or warm air supply ducts are calculated as to their size according to B.t.u., etc., requirements.*

*See Air Conditioning—Furna╌   nd Unit H╌╌t   ╌rican Technical Society, Chicago.

The size of the furnace depends on the number of B.t.u.'s required or upon the total area of all leaders or warm air supply ducts.

If the system is changed so that part of the air is recirculated and part taken from the outside, the size of the furnace must be made larger to meet the increased demands. Where the 100 per cent recirculation principle is used the return air, as it reaches the furnace, is not many degrees below room temperature and thus does not require nearly as much heating as outside air would. Where part outside air and part recirculated air are used in a system, the recirculated and incoming outside air are mixed before reaching the furnace. Adding outside air, from 0° to 45°F., for example, to the recirculated warmer air lowers the recirculated air temperature considerably and thus the furnace must be increased in grate area to supply the increased amount of heat necessary to warm the colder air. Likewise, if all the air is taken from the outside, an even larger furnace is required as there is a great difference between recirculated air and 0°F. outside air. When designing a furnace, the most severe conditions (average) should be assumed.

Residences really do not need the introduction of outside air by means of ducts into the circulation to obtain fresh air. It has been found by actual trials in well-built houses that the amount of infiltration in calm weather was equal to approximately 10 complete air changes per day.* Moderate winds, not to mention brisk winds, add even more. Therefore it seems unnecessary to add fresh air by means of ducts to meet the ventilation needs of the average residence.

Besides fuel and equipment economy, recirculation of air makes possible an increased humidity, especially when the humidifiers employed are not of the most efficient type.

If a residence warm air system contains filters or air washers, the recirculation of air will prove more economical, because in passing through the residence the air does not pick up much dust or dirt and therefore does not put much of a load on such equipment.

Fig. 4 shows a typical warm air furnace equipped to warm and cool the air in addition to filtering out the dust and dirt. During the winter months the furnace is fired and the cooling coil is inoperative. Ducts E and F are the ducts supplying heated air to the rooms. Ducts A and B are the ducts carrying the return air from the rooms

*"Fan Engineering," Buffalo Forge Co.

back to chamber D, which it enters before being filtered. Duct C leads to the outside and can be used to take in fresh outside air as needed. Dampers X can be set manually or by thermostat to control the exact amount of each kind of air that enters the mixing chamber D. Thus the system can be operated entirely recirculating or partially recirculating in various degrees. It is recommended that furnace systems have a provision such as duct C for reasons which are explained a little later in this section.

Cooling Systems. *Residences.* When the type of furnace shown in Fig. 4 is used as a cooling system, the furnace is inoperative but ducts E and F are used nevertheless to distribute cool air.
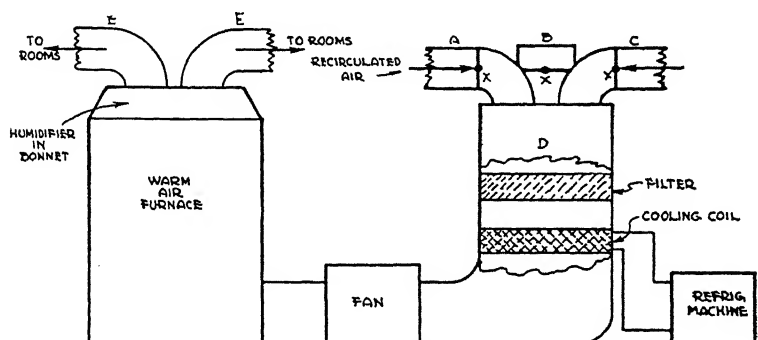
Fig. 4. Warm-Air Furnace and Cooling System for Recirculating and Mixing Air

During very hot weather the refrigerating machine will be operated continuously because of the needs of the coil which cools and dehumidifies the air. It would be economical to recirculate the air, using ducts A and B and having duct C closed. The returned air will still be cooler than air that might be taken in from outside through duct C. Thus the refrigerating machine and coil would not be under nearly the load they would be if duct C were opened to any degree. As duct C is opened the refrigerating load increases, which raises the cost of energy.

On medium warm days some outside air can be used without increasing the load greatly. But as before, the more outside air used the heavier the filter load.

Generally, after sundown, the outside temperature drops a few degrees at least and often several degrees. Then duct C can be partly

or wholly opened, depending on temperature, and thus supply cooler air than the air which is returning from the rooms. Often during the summer the outside air becomes cool enough for comfort after nightfall. In such a case duct C can be fully opened and ducts A and B closed. The refrigeration machine can be inoperative while the fan supplies the outside cool air to the rooms. Duct C, to do this, must be of ample cross-sectional area.

From the foregoing it can be seen that a flexible system, such as is shown in Fig. 4, can be installed and operated economically, especially if automatically controlled. (See Chapter XII.)
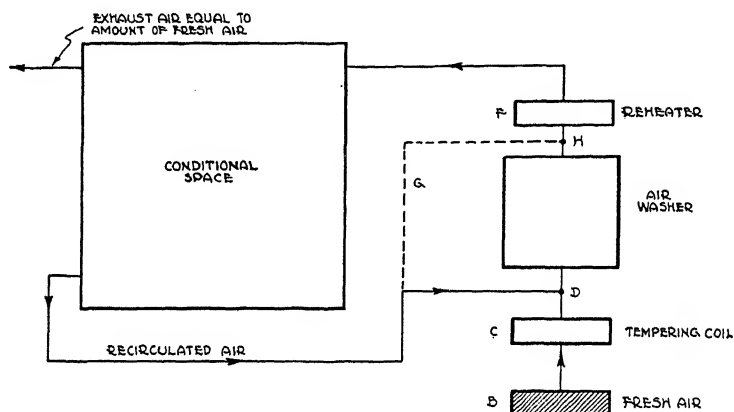
Fig. 5. Diagram for Heating Season Showing Recirculation Plan (Fan not shown)

Larger Buildings. These same principles hold true for larger buildings such as schools where boilers are employed to supply steam or hot water for direct radiation and also for ventilation. Where a quantity of air is being supplied at regular intervals for ventilation it is economical to recirculate a portion of the air for the reason that this air must be heated to room temperatures before being supplied to the rooms. It is understood that to recirculate air in schools, for example, it must be conditioned after each cycle.

If a building is both heated and ventilated by a fan system, it can easily be seen that if more outside air is used more heat is required and if more recirculated air is used less heat is required.

The recirculation of air can be carried on to a point where the humidity in the rooms becomes too high. This should be guarded

against. To avoid the above condition, outside air and recirculated air should be mixed so that the mixture, where an air washer is used, enters the washer at about 45°F. This mixing is taken care of by thermostats which control damper motors in the supply ducts.

Fig. 5 shows a diagram of a heating system used to heat and condition a given space. This system uses heating coils and an air washer for heating, cleaning, and humidifying. The outside or fresh air enters through louvers at B and is tempered in the coil at C so that it enters the air washer well above freezing if no recirculated air is mixed with it during severe weather. The function of coil C, during 0°F. weather, is to heat the fresh air from 0°F. to above freezing. If some recirculated air is mixed with fresh air at D then the coil at C need not heat the fresh air nearly so much, because the recirculated air is not much below 70°F. Mixing slightly heated fresh air with recirculated air therefore cuts down the coil size and the amount of steam needed at C. Further economy can be made by by-passing the recirculated air around the air washer line G and mixing it with the fresh air at H. This would require a smaller washer, less water, and less electrical energy. Coil F reheats as necessary to bring the temperature of the air up to a required standard before entering the conditioned space. Whichever manner of recirculating is used, it brings about a saving in fuel and original cost of boilers and other apparatus. Thermostats are used to control such a system so that it operates entirely automatically. (See Chapter XII.)

During the cooling season the heater coils are inoperative and the air washer is supplied with refrigerated water so that it cools and dehumidifies the air. The same principles regarding the use of outside air hold true for furnace systems, as has already been explained.

By=Pass System. Fig. 6 shows the principle of the by-pass system. This principle is employed in Fig. 5 at G. It is economical to by-pass some of the air, in a cooling system for example, to keep down the required size of the apparatus and save on electrical energy, water, etc. Besides being economical, the by-pass system allows the entering air to be at a desirable temperature. The introduction of very cold air into conditioned spaces is not desirable because of discomfort, etc. Therefore, the cold air is mixed with some recirculated air to bring the temperature to a desirable point. Fig. 6 illustrates how by-passing is done. The air at H can be a mixture of the air

coming from the washer and recirculated air. In this case the return air B is divided at C. Some of it goes through duct K and mixes with the conditioned and cooled air from J and then enters A to form the introduction air before passing to the conditioned space. The balance of the air passes through duct E, joins any incoming outside air from duct F and forms a mixture of these at G. It then enters the air washer where it is cleaned and dehumidified and emerges as conditioned air at J. The various dampers control the mixing automatically or manually.
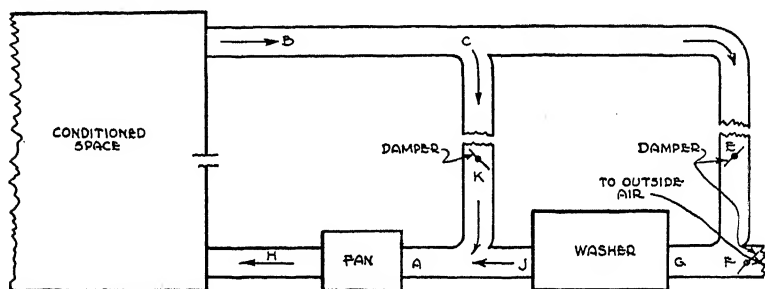
Fig. 6. By-Pass System

It can be seen that the by-pass system has some important benefits in addition to, and combined with, recirculation principles. A little thought will prove that there are numerous combinations possible with systems such as shown in Figs. 5 and 6.

Note: For an example of by-passing in an actual problem, see Example 6 in the section on Designing Cooling Systems, page 430.

*Calculation of Heat Losses and Heat Gains. *Heat Transfer*. During either winter or summer months, if conditions within occupied spaces are kept at comfortable levels, there will be a difference between outside and inside temperatures. In the winter the temperature difference is generally up to 70°F. and during the summer the difference runs as high as 30°F. If, for example, the outside temperature is 0°F. and the inside 70°F., then the difference is 70°F. With such conditions there is bound to be a transfer of heat from the inside to the outside because by natural law heat travels from high to low. The transfer is carried on through walls, windows, doors, roofs, etc.,

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*It is assumed that the reader understands such calculations. The material in this section is merely a review. For full explanation of heat loss and heat gains, see "Furnaces And Unit Heaters" published by American Technical Society, Chicago.

at a rate depending on the conductivity of the materials of which the various structural parts are constructed. Some materials have a higher conductivity than others. The passage of heat is, therefore, at varying rates governed by the materials and also by their thicknesses. In the summer the transfer of heat is from the outside to the inside and practically the same features govern its calculation.

Stated in simple terms, the calculations carried on to ascertain the amount of heat loss from a building are made by multiplying the number of square feet in the area of structural parts, through which losses take place, by the proper coefficient for such a material, and by the temperature difference between interior and exterior spaces. Stated in formula outline this is,

$$H_t = A U(t-t_o) \tag{6}$$

where H_t = B.t.u.'s transmitted per hour through the wall or other section being considered

A = the area in square feet of the wall or other part being considered

$t-t_o$ = the temperature difference between interior and exterior spaces

U = a coefficient which will be explained a little later

Example. An exposed wall has an outside temperature of 0°F. and an inside temperature of 70°F. Assume that the net wall area is 250 square feet and that the value of U is .50. Find the heat loss through the wall in B.t.u.'s.

Solution. We know what the various parts of the formula mean and that t = inside temperature and t_o = outside temperature. Thus

$$t = 70°F.$$
$$t_o = 0°F.$$
$$A = 250$$
$$U = .50$$

By substituting in Formula (6)

$$H_t = 250 \times .50(70° - 0°)$$
$$H_t = 125 \times (70°)$$
$$H_t = 8,750 \text{ B.t.u.'s loss per hour.}$$

Coefficients. When calculating heat losses, we are often able to use a known coefficient, such as U in Formula (6), and quickly determine the losses. But in many cases the value of U (over-all transmission coefficient) is not known and must be calculated before it can be used in Formula (6). Tables 7 and 8 show typical values of U for various types of walls, roofs, glass, doors, etc. Tables 9 and 10

show various k values for typical materials. The values from Table 7 can be substituted for U in Formula (6) but the values in Table 8 must be used in determining U values. This process is explained as follows:

Calculating Coefficients. For a simple, solid wall all of one material, the formula for the over-all coefficient when the wall is x inches thick is:

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{x}{k} + \dfrac{1}{f_o}} \tag{7}$$

If we knew the value of U, we could simply substitute it in Formula (6). But, assuming we do not know the U value it is necessary to calculate it, using Formula (7).

The meaning of the letters in Formula (7) are

f_i = inside surface conductance. This is taken as 1.65. (See Table 9.)

x = thickness of material of which wall is made

k = thermal conductivity. (See Tables 9 and 10.)

f_o = outside surface conductance. This is taken as 6.00. (See Table 9.)

For a wall consisting of several different materials, without any air spaces between them, and having thicknesses of x_1, x_2, x_3, etc., Formula (7) becomes:

Table 7. Transmission Coefficient (U) for Typical Construction Types. Taken from A.S.H.V.E. Guide, 1936

Solid Brick Walls (4 inches face brick—balance common brick)	Thickness in Inches		
	8	12	16
Plain wall—no plaster	.50	.36	.28
½-inch plaster on inside	.46	.34	.27
Plaster on wood lath—furred	.30	.24	.20
Plaster on metal lath—furred	.32	.25	.21
Plaster on ½-inch rigid insulation—furred	.22	.19	.16
Plaster on 1-inch rigid insulation—furred	.16	.14	.13

Table 7—Continued

Concrete Walls	Thickness in Inches			
	6	10	16	20
Plain Wall—no plaster..........................	.79	.62	.48	.41
½-inch plaster on inside......................	.70	.57	.44	.39
Plaster on wood lath—furred......................	.39	.34	.29	.27
Plaster on metal lath—furred......................	.42	.37	.31	.28
Plaster on ½-inch rigid insulation—furred............	.26	.24	.21	.20
Plaster on 1-inch rigid insulation—furred............	.19	.18	.16	.15

Brick Veneer (4 inches) Walls Backed by Hollow Tile 6, 8, 10, and 12 inches	Thickness in Inches			
	10	12	14	16
Plain wall—no interior finish......................	.36	.34	.34	.27
Plaster ½-inch on inside of wall......................	.34	.33	.32	.26
Plaster on wood lath—furred......................	.24	.24	.23	.20
Plaster on metal lath—furred......................	.25	.25	.24	.21
Plaster on ½-inch rigid insulation—furred............	.19	.18	.18	.16

Concrete Floors and Ceilings	Thickness in Inches			
	4	6	8	10
Plain Concrete—no finish..........................	.65	.59	.52	.48
With suspended metal lath plaster ceiling.............	.36	.34	.33	.31
Suspended ceiling—1 inch pine floor.................	.28	.26	.24	.23
Same with ¹³⁄₁₆-inch maple or oak floor over pine	.23	.21	.20	.20

Concrete Floor on Ground	Thickness in Inches			
	4	6	8	10
Plain..	1.06	.91	.79	.70
With 1-inch yellow pine floor......................	.51	.49	.45	.40
With ¹³⁄₁₆-inch maple or oak floor over yellow pine.	.37	.35	.33	.31

Flat Roofs with Built-Up Sheet Roofing	Thickness in Inches		
	2	4	6
Plain concrete...................................	.82	.71	.64
With 1-inch rigid insulation........................	.24	.23	.22
With 2-inch cork..................................	.12	.11	.11
Suspended ceiling—no insulation...................	.41	.39	.36

Wood Construction Walls	Thickness in Inches		
	1	1½	2
Plain..	.48	.37	.33
With 1-inch rigid insulation........................	.21	.19	.16
Suspended ceiling—no insulation...................	.33	.26	.25

Table 7—Continued

Walls and Partitions	*U* Values
Wood siding or clapboard, 2x4 studs, wood lath, plaster	.25
With ½-inch rigid insulation as sheathing	.23
With plaster on metal lath	.26
With plaster on ½-inch rigid insulation	.19
With plaster on 1-inch rigid insulation	.15
With plaster on 1½-inch corkboard	.11
With plaster on metal lath—rock wool fill between studs	.061
With plaster on plaster-board	.25
Shingle Walls, 2x4 studs, wood lath, wood sheathing, plaster	.25
With ½-inch rigid insulation in place of sheathing	.19
With plaster on metal lath	.26
With plaster on plaster board	.25
With plaster on ½-inch rigid insulation	.19
With plaster on 1-inch rigid insulation	.15
With plaster on 1½-inch corkboard	.11
With plaster on metal lath—rock wool fill between studs	.061
Stucco Walls, 2x4 studs, wood sheathing, wood lath and plaster	.30
With plaster on metal lath	.31
With plaster on plaster board	.30
With plaster on ½-inch rigid insulation	.22
With plaster on 1-inch rigid insulation	.16
With plaster on 1½-inch corkboard	.12
With plaster on metal lath—rock wool fill between studs	.064
Brick veneer walls, 4-inch brick, sheathing, 2x4 studs, wood lath and plaster	.27
With plaster on metal lath	.28
With plaster on plaster board	.27
With plaster on ½-inch rigid insulation	.20
With plaster on 1-inch rigid insulation	.15
With plaster on 1½-inch corkboard	.12
With plaster on metal lath—rock wool fill between studs	.062
Partitions, on frame—	
Studding with wood lath and plaster both sides	.34
Studding with metal lath and plaster both sides	.38
Plastered Masonry Partitions	
4-inch hollow tile—plastered both sides	.40
4-inch brick, plastered both sides	.41

Floors and Ceilings

½-inch plaster, wood lath, joists, 1-inch yellow pine floor above	.28
Same, but floor above	.61
¾-inch plaster, metal lath, joists, 1-inch yellow pine floor above	.30
Same, but floor above	.69
Same, with ¹³⁄₁₆-inch maple or oak laid over yellow pine	.24

Pitched Roofs

No ceiling, rafters exposed, wood shingles on wood strips	.48
No ceiling, rafters exposed, asphalt shingles or composition shingles on wood sheathing	.56
Wood lath and plaster ceiling, wood shingles on wood strips	.28
Same with 1-inch flexible insulation	.12

Windows

Single	1.13
Double	.45
Triple	.281

Solid Wood Doors

1 x ²⁵⁄₃₂ inches	.69
1¼ x 1¹⁄₁₆ inches	.59
1½ x 1⁵⁄₁₆ inches	.52
1¾ x 1⅝ inches	.51
2 x 1⅝ inches	.46

Table 8. *Coefficients of Transmission (U) of Bright Aluminum Foil in Various Wall Types

Wall Type	Layers of Foil	Over-All Heat Transmittance
Frame	3	.08
Frame	2	.10
Frame	1	.13
Frame	0	.26
†Brick veneer, 1-inch air space	3	.07
Brick veneer, 1-inch air space	2	.08
Brick veneer, 1-inch air space	1	.12
Brick veneer, 1-inch air space	0	.22
‡Concrete block	3	.08
Concrete block	2	.10
Concrete block	1	.15
Concrete block	0	.32
Stucco	3	.08
Stucco	2	.09
Stucco	1	.14
Stucco	0	.30
Brick veneer, 4-inch, on 8-inch tile	3	.08
Brick veneer, 4-inch, on 8-inch tile	2	.09
Brick veneer, 4-inch, on 8-inch tile	1	.13
Brick veneer, 4-inch, on 8-inch tile	0	.24
Stucco on 8-inch tile	3	.08
Stucco on 8-inch tile	2	.09
Stucco on 8-inch tile	1	.13
Stucco on 8-inch tile	0	.26
‡8-inch concrete walls	3	.09
8-inch concrete walls	2	.10
8-inch concrete walls	1	.16
8-inch concrete walls	0	.36
Brick (8-inch) walls	3	.08
Brick (8-inch) walls	2	.09
Brick (8-inch) walls	1	.14
Brick (8-inch) walls	0	.30

*Data, Courtesy of Alfol Company.
†Air space between sheathing and bricks.
‡Insulation between furring strips. Strips against masonry

Table 9. *Conductivities (k) of Typical Building Materials and Insulators

The coefficients are expressed in B.t.u. per hour per square foot per degree F. per 1 inch thickness, unless otherwise indicated by the thickness or total.

Material		Material	
Brick, common	5.00	Stucco	12.00
Brick, face	9.20	Tile (4 inches)	1.00
Cement mortar	12.00	Tile (6 inches)	.64
Cinder concrete	5.20	Tile (8 inches)	.60
Cinder blocks (8 inches)	.62	Tile (10 inches)	.58
Cinder blocks (12 inches)	.51	Tile (12 inches)	.40
Concrete (typical)	12.00	Hollow tile (4 inches) plaster.	
Concrete blocks (8 inches)	1.00	Both sides	.60
Concrete blocks (12 inches)	.80	Sawdust	.41
Plaster board (½-inch)	2.82	Air Spaces	1.10
Asbestos roof shingles—Total	6.00	Surfaces, still air (f_i)	1.65
Asphalt roof shingles—Total	6.50	Surfaces, 15 m.p.h. (f_o)	6.00
Slate shingles	10.37	1 inch sheathing—paper—Total	.71
Wood shingles—Total	1.28	Yellow pine lap siding—Total	1.28
Plaster—cement	8.00	Yellow pine or fir	.80
Plaster—gypsum—total	3.30	Maple or oak	1.15
Metal lath and plaster—Total	4.40	Plaster board—⅜-inch thick	3.73
Wood lath and plaster—Total	2.50	Balsam wood	.58
Stone (typical)	12.50		

*Taken from A.S.H.V.E. Guide, 1936.

Table 10. Conductivities (*k*) of Typical Insulating Materials by Trade Name

The coefficients are expressed in B.t.u. per hour per square foot per degree Fahrenheit per 1-inch thickness

Insulation	Description	k	Insulation	Description	k
Cabot's Quilt	Quilt form paper covered.......	.25		Rigid type—integral roof insulation, ½-inch.....	.33
Eagle Picher	Wool type—Loose form........	.27		Rigid type—cold storage form, 1-inch...........	.28
	Wool type—Bat form..........	.27		Rigid type—tile B.B..........	.327
	Wool type—Blanket form......	.27		Sealdslab, 1-inch.............	.326
Red Top	Wool type—Bat form........	27		Rigid type—asphalted-roof ½-inch....................	.346
Johns Manville	Wool type—Bat form..........	.27		Fiberock—loose............	.26
	Rigid form, ½-inch.............	.33		Fiberock—granulated..........	.26
	Rigid form, 1-inch.............	.33		Fiberock—bats...............	.26
	Flexible form, ½-inch.........	.27	Alfol	Bright aluminum foil. U values can be found in Table 8.	
	Flexible form, 1-inch..........	.27	Celotex	Rigid type—building board.....	.33
Balsam Wool	Wool type—Blanket form......	.25		Rigid type—plaster backing....	.33
Ruberoid Pipe Covering	85% Magnesia, *100°F........	.425	Dry Zero	Pliable slab form.............	.246
	85% Magnesia, *200°F........	.465	Ozite	Hair blanket type.............	.246
	85% Magnesia, *300°F........	.505	Insulex	Cellular gypsum type..........	.35
	85% Magnesia, *400°F........	.550	Balsa Wood	Wood type..................	.33
	Supercell (14–16 laminations per inch)		Temlok	Rigid type—plaster backing....	.34
	Supercell *100°F..............	.408		Rigid—building board.........	.34
	Supercell *200°F..............	.479	Nu-Wood	Board form..................	.324
	Supercell *300°F.............	.550		Interior finish plank...........	.324
	Supercell *400°F.............	.620		Interior finish wainscot.........	.324
	Watcell *8 plies per inch)			Insulation lath...............	.324
	Watcell *100°F...............	.480		Roof insulation—board........	.35
	Watcell *200°F...............	.555		Wood insulation—board.......	.35
	Watcell *300°F...............	.630		Cold storage insulation........	.35
	Watcell *400°F...............	.705	Thermax	Rigid type...................	.46
	Air Cell (4 plies per inch)		Thermofelt	Felted type..................	.28
	Air Cell *100°F................	.530		Wool type..................	.35
	Air Cell *200°F................	.650	Sil-O-Cel	Powdered diatomaceous earth	.31
	Air Cell *300°F................	.770	Thermofil	Powdered form gypsum........	.52
	Air Cell *400°F................	.890	Eureka	Corkboard type	.32
	*Mean temperature. (The mean of the inner and outer surface temperatures of the insulation.)		Armstrong Cork	Rigid type	
				7 lbs. per cu. ft., 90° mean temp.	.27
				8 lbs. per cu. ft., 90° mean temp.	.28
Masonite	Rigid type...................	.321		9 lbs. per cu. ft., 90° mean temp.	.29
Red Top	Wool type.....	.266		10 lbs. per cu. ft., 90° mean temp.	.30
	Weatherwood board...........	.33		11 lbs. per cu. ft., 90° mean temp.	.31
	Weatherwood lath.............	.33		15 lbs. per cu. ft., 90° mean temp.	.35
	Plank.......................	.33		7 lbs. per cu. ft., 80° mean temp.	.265
	Tile........................	.33		10 lbs. per cu. ft., 80° mean temp.	.295
Burgess Acousti-Pad	Sound-absorbing material for ducts.....................	.246		7 lbs. per cu. ft., 50° mean temp.	.255
Homasote	Rigid type—building board....	.355		8 lbs. per cu. ft., 50° mean temp.	.265
	Rigid type—plaster backing....	.37		9 lbs. per cu. ft., 50° mean temp.	.275
Insulite	Rigid type—building board ¾-inch.............	.327		17 lbs. per cu. ft., 50° mean temp.	.355
	Rigid type—plaster backing....	.327			
	Rigid type—Biltrite sheathing..	.36			
	Rigid type—plank, B-B joints, ½-inch.........	.327			

Note: The basic method of comparing insulating materials is by comparison of their "Conductivity." However, as these materials are generally available in several thicknesses, and are used in combination with other materials, it is the measurement of the actual installation of these materials which provides a correct measure of wall, roof, floor, or partition as to its heat-transmitting properties.

Thus no accurate comparisons can be made between items in Table 10.

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{x_1}{k_1} + \dfrac{x_2}{k_2} + \dfrac{x_s}{k_3} + \dfrac{1}{f_o}} \qquad (8)$$

The meanings of the letters are exactly the same as explained for Formula (7). In Formula (8) we use x_1, x_2, and x_3 to denote thickness of the three materials making up an assumed wall.
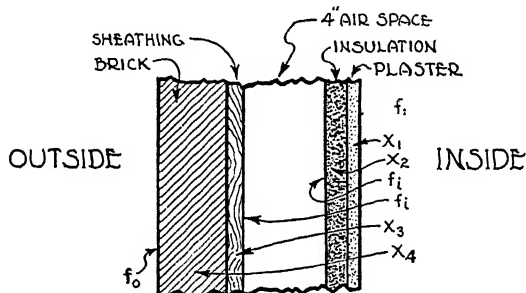

Fig. 7. A Typical Wall

If a wall, without air spaces, consisted of say, 6 or 8 different materials, then Formula (8) would be extended so that it would contain from $\dfrac{x_1}{k_1}$ to $\dfrac{x_6}{k_6}$ or $\dfrac{x_8}{k_8}$, etc.

Air Spaces. Most walls, roofs, etc., have air spaces within them as evidenced by the space between wall studs and roof rafters. These spaces are practically dead air spaces because of the plaster and siding on the walls. Most air spaces are over $1\frac{1}{2}$ inches in width so the following method can be used generally. Fig. 7 shows a wall composed of two materials of x_1 and x_2 thickness. There is a 4-inch air space. To apply Formula (8) to such a wall the formula is changed to

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{x_1}{k_1} + \dfrac{1}{f_i} + \dfrac{1}{f_i} + \dfrac{x_2}{k_2} + \dfrac{1}{f_o}} \qquad (9)$$

The $\dfrac{1}{f_i}$ value is applied to all surfaces where there is no wind velocity. This means all inside surfaces and all air space surfaces as shown in Fig. 7. The $\dfrac{1}{f_o}$ value is used on all outside surfaces. If any

wall or roof, etc., contained more than one air space, then these values would be used for each space.

Example. Figure the coefficient of transmission U of an 8-inch brick wall having plaster $\frac{1}{2}$-inch thick applied directly to the inside surface of the wall without lathing. The outside course of brick is face brick and the inside common brick. Wind velocity is assumed, as usual, at 15 miles per hour. Thickness of each brick course is 4 inches. The inside and outside surface coefficients are assumed to be 1.65 and 6.00, respectively.

The solution of such a problem requires one of the basic formulas, and upon studying the formulas we see that formula (8) fits the conditions of our problem; a solid wall (no air space) made up of three different materials.

First write formula (8)

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{x_1}{k_1} + \dfrac{x_2}{k_2} + \dfrac{x_3}{k_3} + \dfrac{1}{f_o}}$$

now substitute these values in the formula

$$\frac{1}{f_i} = \frac{1}{1.65}$$

$$\frac{x_1}{k_1} = \frac{4.0}{9.20}$$

$$\frac{x_2}{k_2} = \frac{4.0}{5.0} \qquad \text{All of these } k \text{ values can be found in Table 9}$$

$$\frac{x_3}{k_3} = \frac{0.5}{3.3}$$

$$\frac{1}{f_o} = \frac{1}{6.0}$$

We know that $f_i = 1.65$ because the problem states that the inside surface coefficient is 1.65. The x_1 and k_1 represent thickness and coefficient for the face brick. The thickness being 4 inches, then $x_1 = 4.0$. Table 9 shows that the coefficient for face brick is 9.20. So $\frac{x_1}{k_1} = \frac{4}{9.20}$. The second x and second k represent the thickness and coefficient for common brick. This is 4 inches wide, so $x_2 = 4.0$. Table 9 shows that the coefficient for common brick is 5.0, so $k_2 = 5.0$. The third x and third k represent the thickness and the coefficient of plaster. The plaster thickness is $\frac{1}{2}$ inch, so $x_3 = 0.5$. Table 9 shows that plaster has a coefficient of 3.3, so $k_3 = 3.3$. The outside surface coefficient was given as 6.00, so $f_o = 6.00$.

Thus the formula becomes:

$$U = \frac{1}{\dfrac{1}{1.65} + \dfrac{4.0}{9.20} + \dfrac{4.0}{5.0} + \dfrac{0.5}{3.3} + \dfrac{1}{6.0}} \qquad \textit{Ans. } U = 0.46$$

Note: There are two methods of solving the above equation. Method 1: it is assumed that the reader understands the procedure of adding fractions, which is necessary before the denominator in the formula can be added. Find a least

common denominator, express all fractions in terms of this denominator and add. Then divide 1 by the result. Method 2: the various fractions can be changed to decimals by dividing their numerators by their denominators. The quotients then become denominators under the numerator of 1. Then divide 1 by the total of the denominators to find U.

The answer for this example is 0.46 B.t.u. per hour per square foot per degree F. difference in temperature between the air on the two sides.

Combined Coefficients. It often happens that the attic space in residences is unheated, and the roofs are generally pitched to some degree. When heat loss is to be calculated where such ordinary conditions are met, the combined coefficients for roof and ceiling are considered in one formula. This saves time and tends to give more accurate results.

$$U = \frac{U_r \times U_{ce}}{n \times U_r + U_{ce}} \qquad (10)$$

where

$U_r =$ coefficient of transmission of the roof

$U_{ce} =$ coefficient of transmission of ceiling

$n =$ ratio of the roof area to the area of ceiling

This formula assumes the calculation per square foot of area.

To use this formula for good results a correction factor must be kept in mind. The amount of heat transferred through an air space is proportional to the difference of the fourth powers of the absolute temperatures of the surfaces enclosing the air space. Thus a greater amount of heat is absorbed or emitted by radiation by the surfaces enclosing an unheated attic than by the surfaces of a wall or ceiling in a room under still air conditions, where the surrounding objects are only slightly higher in temperature than the inside surfaces of walls, etc. To explain further, the average coefficient of a surface in still air is 1.65 B.t.u. per hour per square foot per degree F. compared to the average coefficient of an air space in an outside wall of 1.10 B.t.u. per hour per square foot per degree F. difference between the two areas. An air space coefficient of 1.10 is about the same as a surface coefficient of 2.20 for each of the two surfaces enclosing the air space when the over-all transmission is computed by using the coefficients of the two surfaces enclosing the space, instead of the coefficient of the air space itself. So the coefficients should be increased to allow for the amount of heat transferred and a coefficient of 2.20 may be used for each area or surface.

If there are no dormers, windows, or vertical wall areas, combined coefficients may be used to determine heat loss, but the coefficients should be multiplied by roof area and not ceiling area. If the attic space has windows, ventilators, etc., which tend to keep the enclosed air at or near the outside temperatures, the roof should be left out of the calculations and only the ceiling and floor construction and area be taken into consideration. Then coefficients of Tables 8 or 9 are to be used. In case there are no dormers, windows, etc., the attic space temperature can be assumed as being the average between the inside and outside temperatures.

From this discussion it is evident that ordinary good judgment is all that is necessary and where very good results are required, maximum conditions should always be assumed in order to bring about some degree of safety factor. When insulation enters into the calculations, even more care must be exercised because of the added cost involved in the structure. Insulation, while being an agent to promote comfortable occupancy of a residence, also aims to effect economy of fuel and power. Therefore, if design calculations were inaccurate, almost a total loss in both comfort and economy factors might be experienced.

When heat losses are being calculated for walls or partitions between two rooms or areas, one being heated and the other not, we can generally assume the unheated space to be at a temperature of 32°F. if it is closed and not open to the weather. If such a room or space is open in any way, as by windows or door left open or by ventilators, then we assume its temperature to be the same as the outside temperature.

In figuring losses through first floors, the basement is assumed, in extreme cases, to be 32°F. The basement, however, is usually only a few degrees cooler in the winter than the first floor areas because of the heating equipment, pipe runs, etc., so that the designer may assume its temperature as being considerably above 32°F. if economy of insulation is a basic principle in the construction, etc.

Walls or partitions next to an entry way or vestibule are calculated as to heat loss by assuming a temperature of 32°F. unless the entry or vestibule is likely to be open a great deal, in which case the outside temperature should be assumed for it.

Leakage through Windows. Table 11 gives the amounts of in-

filtration for various types of windows, expressed in cubic feet of crack per hour.

The total length of crack for a double-hung window is obtained by adding the lengths of the two vertical sides, the top, the bottom, and the meeting rail. The table shows results for poorly fitted windows as well as for well constructed ones, and the designer must assure himself of good construction in new work if he is to use latter values. For old structures, the values for poorly fitted windows are used unless considerable work is to be done to repair and tighten up.

Table II. *Infiltration through Windows

Expressed in cubic feet per foot of crack per hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]...	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]..	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weather-stripped, $1/16$-inch crack and $3/64$-inch clearance[c]. Includes wood frame leakage[d]....	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weather-stripped, $3/32$-inch crack and $3/32$-inch clearance[e]. Includes wood frame leakage[d]...	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted,[g] $1/16$-inch crack	52	108	176	244	304	372
	Architectural projected,[h] $3/64$-inch crack	20	52	88	116	152	208
	Residential casement,[i] $1/32$-inch crack	14	32	52	76	100	128
	Heavy casement section, projected,[j] $1/32$-inch crack	8	24	38	54	72	96
Hollow metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

[c]The fit of the average double-hung wood window was determined as $1/16$-inch crack and $3/64$-inch clearance, by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction, assuming a 50 per cent efficiency of frame calking.

[e]A $3/32$-inch crack and clearance represents a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Vents may swing in or out and are balanced on side arms.

[i]Of same design and section as that residential casement but of lighter weight.

[j]Made of heavy section. Vents that swing in or out and stay set at any degree of opening.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With $3/64$-inch crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

*Courtesy A.S.H.V.E. Guide, 1936.

Table 11 shows infiltration values for *unlocked* windows, as tests by A.S.H.V.E. have shown that the leakage for locked windows is greater than for unlocked. In order that low infiltration values may be applied to them, windows must be snug fitting, free from warping, well puttied, well painted, and have frames in equally good condition. Any deviation from these specifications should place the window in the poorly fitted class to avoid costly errors in heat-loss calculations. Windows in existing buildings should be examined and rigid specifications insisted upon in new structures.

Casement windows of wood construction are in the same class as a well-fitted double-hung window. If metal pivoted sash is used the total crack is considered to be equal to the perimeter (distance around) of that portion of the sash that is movable. Steel windows have so little leakage that it can be neglected if good workmanship is assured; otherwise the infiltration values equal to industrial pivoted sash in Table 11 should be used.

Leakage through Doors. If a door is well fitted (this means properly tight and having a stop) the same values per foot of crack may be used as for a poorly fitted double-hung window. If weatherstripping is used values may be cut 50 per cent. To merit this rating, doors should not be warped or cracked. If a door has glass in it, the glass should be figured the same as a well-fitted double-hung window, with some cut in values if well fitted and not to be opened. If a door is being opened and closed considerably, its infiltration value should be multiplied by three or four or more, according to how much it is open.

Heat Equivalent of Air Entering by Infiltration. As cold outside air enters a room by infiltration it must be heated. The heat required to accomplish this is calculated by Formula (11).

$$H_i = 0.24 Q d (t - t_o) \tag{11}$$

where

$H_i =$ B.t.u. per hour required for heating air leaking into the building from outside temperature, t_o, to inside temperature, t.

$Q =$ cubic feet of air entering per hour at inside temperature, t.

$d =$ density (pounds per cubic foot) of air at inside temperature, t.

$t =$ inside temperature at proper level.

t_o = outside air temperature for which heating system is designed.

0.24 = specific heat of air.

It is accurate enough to take $d = 0.075$ pound, in which case the equation reduces to

$$H_i = 0.018Q(t - t_o) \tag{12}$$

PRACTICE PROBLEMS

1. A room contains three plain double-hung windows 2 feet 8 inches by 5 feet 6 inches, with $\frac{1}{16}$-inch crack and $\frac{3}{64}$-inch clearance. Assume wind velocity of 20 miles per hour and a temperature difference of 75°F. Calculate maximum heat loss due to infiltration. *Ans.* 4,561 B.t.u. per hour

2. What will be the infiltration through air-dried end and side-matched sheathing for wind at 15 miles per hour? *Ans.* 50 c.f.h. per square foot of wall

3. Using an infiltration figure of 59.3 cubic feet per foot of crack per hour, what will be the heat requirement in a building with total crack (all windows and doors) of 180 feet, if the wind velocity is 15 miles per hour, the outside temperature 0°F., and the inside temperature 70°F. Use the method of solution where half the amount of crack is used and Formula (12). *Ans.* 6,724.6 B.t.u.

4. A solid 12-inch common brick wall is finished on the inside with $\frac{1}{2}$-inch insulation plaster base, and $\frac{1}{2}$ inch of plaster; the plaster base is furred 1 inch from the brick; k for insulating material is .34. Calculate the over-all coefficient U. Assume $f_i = 1.65$ and $f_o = 6.00$ and that mean temperature is 40°F. This latter item is used in determining the value of a for air space. Value of k for brick is found in Table 9. *Ans.* $U = 0.175$

5. Assume a 13-inch brick wall having 1 inch of plaster on the inside surface. Calculate the value of U. Hint: Use Table 9 to find k values for brick and plaster. Find values of f and f_o, assuming 15 miles per hour wind as is generally assumed when not definitely given. *Ans.* $U = 0.280$

6. Assume a wall composed of 4 inches of face brick, sheathing, studs, a 1-inch cork board, and $\frac{1}{2}$ inch of plaster. The air space is between the 2x4 studs. Assume wind at 15 miles per hour. Calculate value of U for such a wall. *Ans.* $U = 0.139$

7. Assume the same wall as in Problem 6 except that wool has been put between the studs to fill up the space entirely. Calculate U. No answer is given, but the U value should be less than in Problem 6.

8. Assume that the wall of Problem 6 has ordinary lath (wood) and plaster in place of cork and plaster; otherwise wall is the same as in Problem 6. Calculate U.

Calculating Heat Losses. The calculation of heat losses from a building (heating load) is done considering the following items: (1) inside temperature, (2) outside temperature, (3) heat transmission, (4) coefficients, (5) transmission losses, (6) infiltration, (7) total heat loss, (8) wind consideration.

1. *Inside Air Temperature.* (See Table 12.) Air temperature should be taken at the breathing line, and about 36 inches from outside walls. The humidity should be taken into account, because an occupant of the room can feel cool or warm at 70°F., depending on the relative humidity. In most cases effective temperatures are considered—the ratio of dry-bulb temperature to relative humidity. Thus if a dry-bulb temperature of 70°F. and a relative humidity of 45 per cent obtained, the effective temperature would be 65.8°F. This effective temperature is regarded as best for people who are more or less sedentary.

If ceilings are less than 20 feet high, the temperature (F.) increases 2 per cent with each foot of distance above the breathing line. Thus, the temperature at a point 3 feet above the breathing line, if the breathing line temperature is 70°F., will be

$$(1.00+3\times.02)70° = 74.2°F.$$

Table 12. Inside Air Temperatures

Hospitals
Private rooms.. 70–72
Wards.. 68
Kitchens... 68
Theatres
Seating spaces... 68–72
Hotels
Bedrooms... 70
Dance Floors... 65–68
Homes.. 70–72
Stores... 65–68
Classrooms... 70–72

2. *Outside Air Temperature.* This temperature will be determined from existing records, which can be secured from local weather bureaus. The minimum temperature should be ascertained. If temperatures should fall slightly below the previous minimum, no harm will be done, as such exceptionally low temperatures last but a day or two at a time and the building in question will probably have stored enough heat to balance the condition. For example, if the lowest recorded temperature for Chicago is 23°F. below zero, we could use as a design temperature 10° or 15° above −23°F. Generally, we would use a temperature 15°F. above the −23°F., that is, about 8° below zero, because the −23°F. figure is no doubt a record of long standing, which may seldom be equaled and then only for a few hours at a time.

3. *Heat Transmission.* Calculate the heat transmission coefficients as explained for all walls, roofs, glass, etc.

4. *Coefficients.* From scaled drawings or by measuring existing buildings, determine areas of walls, roofs, glass, etc., which bound the heated spaces.

5. *Transmission Losses.* Using the information gathered in the foregoing items and the methods explained, calculate the transmission losses for all walls, roofs, glass, etc.

6. *Infiltration.* Using the methods explained, calculate infiltration of cold air around cracks, etc.

7. *Total Heat Losses.* The sum of the heat losses calculated in paragraph 5 and equivalent of cold air obtained in paragraph 6 is the total heat loss.

8. *Wind Considerations.* The heat loss due to wind varies. In a poorly constructed building infiltration as well as loss of heat by transmission would be materially increased during a strong wind. Even well constructed buildings suffer to some extent from the same cause. There are no exact rules governing such conditions, so engineers generally include extra factors in their calculations for transmission and infiltration. If a designer knows a building is subject to higher than normal winds, or that its structure is not wholly capable of resisting normal winds, he can increase the factor f_o. Ordinarily an average wind velocity of 15 miles per hour is assumed. Tall buildings require special infiltration adjustment calculations.

*Example. In the following is presented a typical example of the method of figuring heat losses. A factory building is used as an example. The reader should digest this carefully and thoroughly.

(1) Location. . . . . Philadelphia, Pa.

(2) Lowest outside temperature. (Table 13). . . . . $-6°$F.

(3) Base temperature: In this example a design temperature 10° above lowest on record, instead of 15°, is used. Hence the base temperature $= (-6+10) = +4°$F.

(4) Prevailing wind (Dec., Jan., Feb.). . . . . Northwest

(5) Breathing-line temperature (5 feet from floor). . . . . 60°F.

(6) Inside Air Temperature at Roof: The air temperature just below roof is higher than at the breathing line. Height of roof is 16 feet, or it is $16-5 = 11$ feet above breathing line. Allowing 2 per cent per foot above 5 feet, or $2 \times 11 = 22$ per cent, makes the temperature of the air under the roof $= 1.22 \times 60° = 73.2°$F.

(7) Inside Temperature at Walls: The air temperature at the mean height of the walls is greater than at the breathing line. The mean height of the walls is 8 feet and allowing 2 per cent per foot above 5 feet, the average mean temperature of the walls is $1.06 \times 60° = 63.6°$F. By similar assumptions and calculations, the mean temperature of glass will be found to be 64.2°F.; doors, 61.2°F.

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*Courtesy A.S.H.V.E. Guide, 1936.

Table 13. Average Maximum Design Dry=Bulb Temperatures, Design
Wet=Bulb Temperatures, Wind Velocities, and Wind
Directions for June, July, August, and September

State	City	Average Maximum Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity m.p.h.	Prevailing Summer Wind Direction
Ala.	Birmingham	93	77	5.2	S
Ariz.	Phoenix	110	77	6.0	W
Ark.	Little Rock	95	77	7.0	NE
Calif.	Los Angeles	88	70	6.0	SW
	San Francisco	85	68	11.0	SW
Col.	Denver	90	64	6.8	S
Conn.	New Haven	88	74	7.3	S
D.C.	Washington	95	78	6.2	S
Fla.	Jacksonville	94	78	8.7	SW
Ga.	Atlanta	91	75	7.3	NW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
Ind.	Indianapolis	90	73	9.0	SW
Iowa	Des Moines	92	74	6.6	SW
Ky.	Louisville	94	75	8.0	SW
La.	New Orleans	94	79	7.0	SW
Maine	Portland	85	71	7.3	S
Md.	Baltimore	93	76	6.9	SW
Mass.	Boston	88	73	9.2	SW
Mich.	Detroit	93	73	10.3	SW
Minn.	Minneapolis	84	72	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	92	75	9.5	S
Mont.	Helena	87	63	7.3	SW
Neb.	Lincoln	93	74	9.3	S
Nev.	Reno	93	64	7.4	W
N.J.	Trenton	95	76	10.0	SW
N.Y.	Albany	90	74	7.1	S
	New York	95	75	12.9	SW
N.M.	Santa Fe	87	63	6.5	SE
N.C.	Asheville	87	72	5.6	SE
N.Dak.	Bismarck	88	69	8.8	NW
Ohio	Cleveland	95	73	9.9	S
	Cincinnati	95	78	6.6	SW
Okla.	Oklahoma City	96	76	10.1	S
Ore.	Portland	83	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
R.I.	Providence	85	73	10.0	NW
S.C.	Charleston	94	80	9.9	SW
Tenn.	Chattanooga	94	76	6.5	SW
Texas	Dallas	99	76	9.4	S
	San Antonio	100	78	7.4	SE
	El Paso	98	69	6.9	E
Utah	Salt Lake City	95	67	8.2	SE
Vt.	Burlington	85	71	8.9	S
Va.	Norfolk	91	76	10.9	S
Wash.	Seattle	83	61	7.9	S
	Spokane	89	63	6.5	SW
W.Va.	Parkersburg	90	74	5.3	SE
Wis.	Madison	89	73	8.1	SW
	Milwaukee	93	74	10.4	S
Wyo.	Cheyenne	85	62	9.2	S

(8) Average Wind Velocity (Table 13)......................11.0 m.p.h.
(9) Over-all Dimensions ....120×50×16 feet
(10) Construction:

Walls: 12-inch brick, with ½-inch plaster applied directly to inside
surface.

Roof: 3-inch stone concrete and built-up roofing.

Floor: 5-inch stone concrete on 3-inch cinder concrete on dirt.

Doors: One 12×12-foot wood door (2 inches thick) at each end.

Windows: Fifteen, 9×4-foot single glass, double-hung windows on
each side.

(11) Transmission Coefficients:

Walls. . . . . $U = 0.34$

Roof . . . . $U = 0.77$

Floor. . . . . $U = 0.63$

Doors. . . . . $U = 0.46$

Windows. . . . . $U = 1.13$

(12) Infiltration Coefficients:

Windows: Average windows, non-weather-stripped, $\frac{1}{16}$-inch crack and $\frac{3}{64}$-inch clearance. The leakage per foot of crack for an 11-mile wind velocity is 25.0 cubic feet per hour. (Determined by interpolation of Table 11.) The heat equivalent per hour per degree Fahrenheit per foot of crack is $25.0 \times .018 = 0.45$ B.t.u.

Doors: Assume infiltration loss through door crack twice that of windows, or $2 \times 0.45 = 0.90$ B.t.u. per degree Fahrenheit per foot of crack.

Walls: A plastered wall allows so little infiltration that in this problem it may be neglected.

(13) Calculations: See calculation sheet.

Calculation Sheet Showing Method of Estimating Heat Losses of Building

Part of Building	Width in Feet	Height in Feet	Net Surface Area or Crack Length	Coefficient	Temp. Diff.	Total B t.u.
North wall:						
Brick, $\frac{1}{2}$-in. plaster.	50	16	656	0.34	59.6	13,293
Doors (2-in. wood).	12	12	144	0.46	57.2	3,789
$\frac{1}{8}$ in. Crack	1 pair doors		60	0.90	57.2	1,544[a]
West wall:						
Brick, $\frac{1}{2}$-in. plaster.	120	16	1380	0.34	59.6	27,964
Glass (single).	15×4	9	540	1.13	60.2	36,734
$\frac{1}{16}$ in. crack.	Double hung windows (15)		450	0.45	60.2	6,095[a]
South wall.	Same as north wall					18,626
East wall.	Same as west wall					70,793
Roof, 3-in. concrete and slag-surfaced built-up roofing.	50	120	6000	0.77	69.2	319,704
Floor, 5-in. stone concrete on 3-in. cinder concrete.	50	120	6000	0.63	5[b]	18,900
Grand Total of heat required for building in B.t.u. per hour						517,442

[a]This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack is used in computing infiltration for each side and each end of building. Thus, $(60 \times .90 \times 57.2) \div 2 = 1,544$ and $(450 \times .45 \times 60.2) \div 2 = 6,095$.

[b]A 5-degree temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground.

Calculating Heat Gains. The calculation of heat gains (cooling load) is outlined and illustrated in the section devoted to Designing Cooling Systems.

The transmission loads are calculated by using the following formula

$$H_t = A U(t_o - t) \tag{13}$$

Table 14. *Solar Intensity

SUN TIME·AUGUST 1.
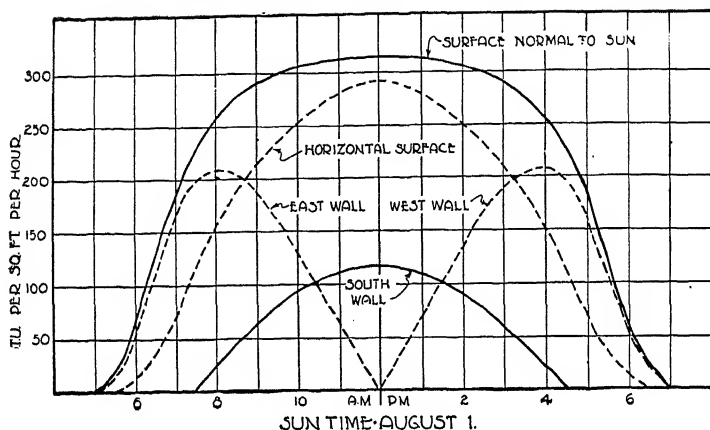
Table 15. *Allowance for Solar Radiation on Roofs and Walls

Approximate number of degrees to add to dry-bulb temperature for different types of surfaces

Type of Surface	Black	Red Brick or Tile	Aluminum Paint
Roof, horizontal	45	30	15
East or west wall	30	20	10
South wall	15	10	5

Table 16. *Solar Radiation Transmitted through Windows

Description	Per Cent Delivered to Room
Bare window glass	96
Canvas awning	28
Inside shade, fully drawn	45
Inside shade, one-half drawn	67
Inside Venetian blind, fully covering window	58
Outside Venetian blind, fully covering window	21

Table 17. *Heat Gain Due to Various Devices, B.t.u. per Hour

Device	B.t.u. per Hour
Lights and electric appliances, per kw	3,415
Motors, 1/10 horsepower	255
Motors, 1 horsepower	2,546
Restaurant coffee urns, 10-gallon capacity	16,000
Dish warmers per 10 square feet of shelf	6,000
Restaurant range—4 burners and oven	100,000
Residence gas range	
Giant burner	12,000
Medium burner	9,000
Oven, per cubic foot of space	1,000
Pilot	250
Electric range	
Small burner, 100 to 1350 watts, per kw	3,415 to 4,600
Large burner, 1700 to 2200 watts, per kw	5,800 to 7,500
Oven, 2000 to 3000 watts, per kw	6,830 to 10,245
Appliance connection, 660 watts	2,250
Warming compartment, 300 watts	1,025

*Courtesy A.S.H.V.E. Guide, 1936.

Outside temperatures are not to be the highest on record because such temperatures seldom occur. Table 13 shows typical temperatures to use as "design temperatures."

Radiation through glass has to do with the amount of energy received in B.t.u. per square foot per hour during the day: by a surface normal to sun's rays, by a horizontal surface, and by east, west, and south walls. This is shown in Table 14.

Table 15 gives generally used allowances for solar radiation on roofs and walls. These allowances are the number of degrees to add to dry-bulb temperatures in computing heat gains.

Table 16 shows solar radiation transmitted through bare and shaded windows. The figures given are the per cents delivered to the rooms. It can be seen that shades help greatly to reduce the load.

In cases where there is an unventilated attic space between roof and ceiling, assume that the temperature between ceiling and roof is 30°F. above the outside temperature, and 10°F. above for an attic having forced ventilation.

Table 17 shows the heat gain due to service applications.

PRACTICE PROBLEMS

1. Given relative humidity of 50 per cent and wet-bulb temperature of 60°F. Find dry-bulb temperature and dew point.

2. Given wet-bulb temperature of 55°F. and dew point of 50°F. Find dry-bulb temperature and relative humidity.

3. Given relative humidity of 40 per cent and dew point of 40°F. Find dry-bulb temperature and wet-bulb temperature.

4. Given dew point of 70°F. and dry bulb of 80°F. Find the relative humidity if the dry-bulb temperature is increased to 90°F., the dew point remaining constant.

5. Given dry-bulb temperature of 70°F. and wet-bulb temperature of 60°F. Find the resulting relative humidity when air is heated to 80°F. (dry bulb) without any increase in moisture content.

6. Given dry-bulb temperature of 80°F. and wet-bulb temperature of 70°F. Find the vapor pressure.

7. Given relative humidity of 20 per cent and dry-bulb temperature of 85°F. Find wet-bulb temperature, dew point temperature, and vapor pressure.

8. Given air at dry-bulb temperature 70°F. Find grains of moisture per cubic foot of air when saturated at this temperature.

9. Given saturated air at 70°F., find volume in cubic feet per pound.

10. Having air at a wet-bulb temperature of 70°F. and air at a wet-bulb temperature of 61°F., find the difference in total heat between a mixture containing one pound of dry air and the quantity of moisture present at the wet-bulb temperature of 70°F., and a mixture containing one pound of dry air and the quantity of moisture present at the wet-bulb temperature of 61°F.
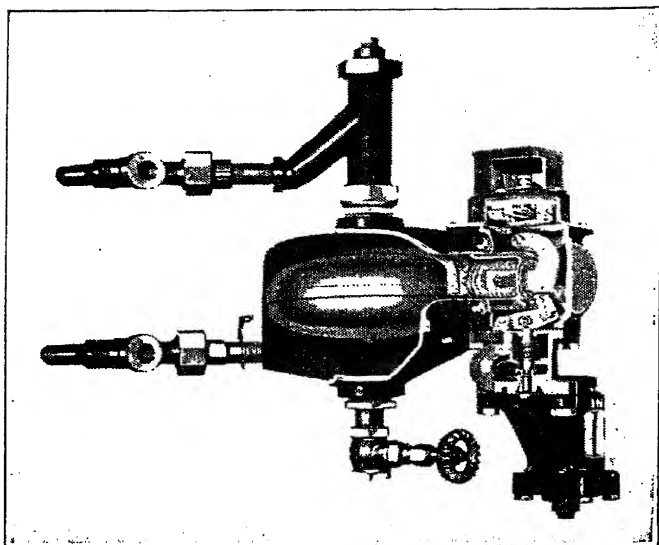
SAFETY FEEDER AND LOW WATER CUT-OFF
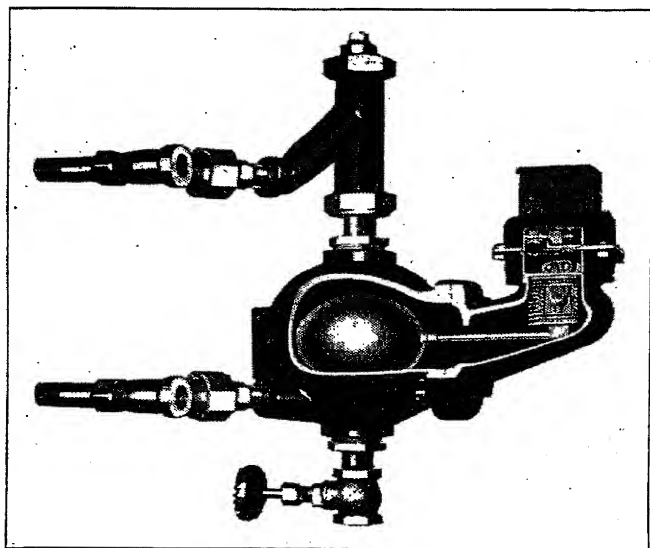
QUICK HOOK-UP LOW WATER CUT-OFF
Courtesy of McDonnell & Miller, Chicago

CHAPTER III

COMFORT STANDARDS

Opinions vary to such an extent in regard to comfort standards that no completely uniform specifications have been compiled since air conditioning became popular. There are certain factors, however, which must be considered in any air-conditioning design and these will be discussed here.

Dust. Particles of dust are always present in the atmosphere. Dust has several useful functions, among them the decimation of light and the condensation of moisture. In cities the amount of dust is too great, however, and it becomes a menace to comfort and health. Some dust particles have sharp-pointed edges which are invisible to the naked eye yet harmful to the lungs, and most forms of dust act as carriers of disease germs. Therefore, it is important to eliminate dust from air-conditioned interior spaces. The methods for removing dust from the air are discussed elsewhere in this book.

Ozone. Ozone is a form of oxygen molecule. It is active as an oxidizing agent because it is unstable in character. Ozone is used to destroy bacteria in the air, to offset fumes, and to mask odors. Ozone must be highly concentrated to make its use of value. The highest permissible concentration for air conditioning is 0.1 part per million parts of air. Ozone is generated by an electrical generator which is made especially for use in connection with air-conditioning systems.

Odors. Odors affect the olfactory nerve, which is the organ of smell. Odors may be gaseous, mist, or solid matter. When an odor, in one of these three forms, is impinged upon the mucous covering of the olfactory nerve, the odor is dissolved, the resulting solution stimulates the nerve, and a person "smells" the odor.

In general, odors are not harmful to health—aside from causing temporary sickness or loss of appetite. However, they make an enclosure unpleasant and air-conditioning apparatus should, so far as possible, eliminate odors.

Air Motion. Some motion of the air is desirable at all times. Stagnant air is neither comfortable nor healthful, no matter how

pure it is, or what its temperature and humidity condition. From available data, it seems desirable that 5 feet per minute should be the minimum during the heating season and 50 feet per minute the maximum during the cooling season. The air motion and the turbulence may be measured with a Kata thermometer. Smoke puffs also may be used in combination with a stop watch to estimate roughly the velocity of the air movement. This latter method has the advantage of presenting a visualization of the air currents and their direction.

The total amount of air circulated in any system is the sum of the quantity taken from outside plus the quantity of air recirculated. The more modern systems of ventilation maintain a set minimum of outside air and vary the ratio of outside air and recirculated air, maintaining correct temperature conditions within the building. This applies to a heating and ventilating system.

When considering a cooling and dehumidifying system for summer air-conditioning work, it is desirable to plan to recirculate as great a portion of the air as practical, because of the relatively high cost of cooling. There will always be a considerable quantity of outside air coming into a building by infiltration through walls and around windows and from opening of doors. On many installations, such as residences, uncrowded office buildings, and similar places (where the number of occupants is low compared with the occupied volume) this infiltration supplies sufficient fresh air. Accordingly, equipment should be planned for a totally recirculated job.

*Effective Temperature. From the foregoing it is evident that the three factors, (1) temperature, (2) humidity, and (3) air motion are closely interrelated in their effects upon comfort and health. The atmospheric condition at any time may be such that these various influences may act in opposite directions, so it is in the combined or net effect that we are interested. This net effect is "effective temperatures." The numerical value of the Effective Temperature Scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

The American Society of Heating and Ventilating Engineers, after examination of all available data, has agreed upon two tables of effective temperatures. These tables are based upon dry-bulb

* Courtesy of Bryant Air Conditioning Manual

temperatures and the relative humidity for still air, for persons normally clothed and slightly active. Still air refers to an air movement not in excess of 25 feet per minute. The first of these two tables, Table 18, is for effective temperatures ranging from 64°F. to
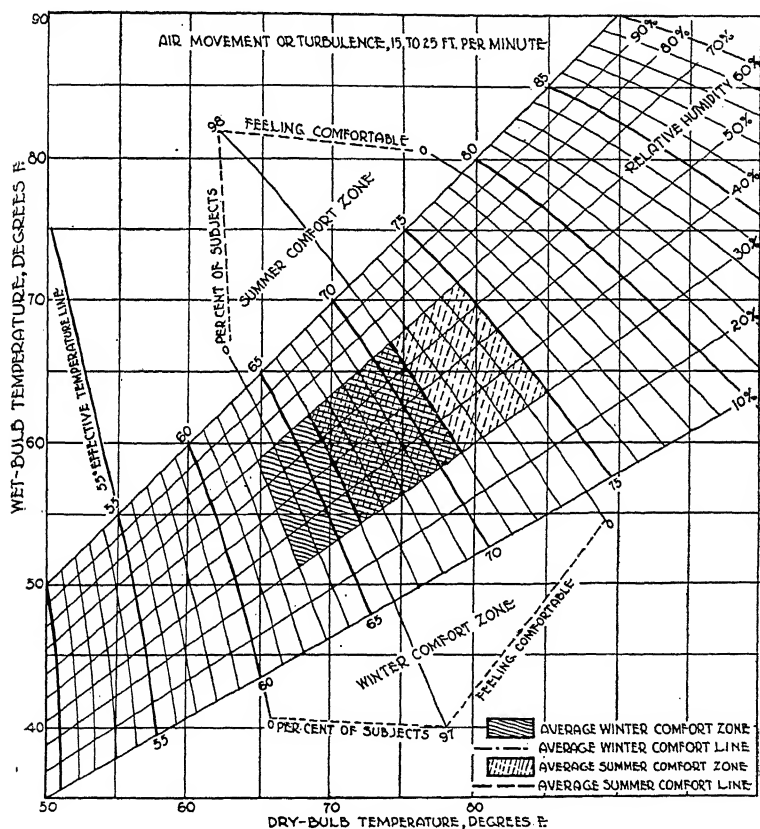
Fig. 8. Comfort Chart
Courtesy of American Society of Heating and Ventilating Engineers Guide, 1937

69°F., and is for use when heating and humidification are required. The second of these tables, Table 19, is for effective temperatures ranging from 69°F. to 73°F. and is for use when either cooling or dehumidification is required.

The American Society of Heating and Ventilating Engineers Research Laboratory has prepared also a "Comfort Chart," Fig. 8.

This was plotted with the wet-bulb temperature as the ordinate and the dry-bulb temperature as the abscissa. The straight oblique lines represent relative humidity and the curved oblique lines represent effective temperatures. The comfort zones are designated by shading on the chart—one for summer and one for winter.

Example for Comfort Chart Use. Using Fig. 8 and assuming a dry-bulb temperature of 85°F. and a wet-bulb temperature of 66°F. (a measure of relative humidity), the effective temperature is 76°F. Find 85°F. along the bottom line of the chart. Follow this line upward until it intersects an imaginary line drawn from 66°F. on the left-hand line of the chart. The two lines intersect on the 76°F. effective

85° F. DRY-BULB
AND
30% RELATIVE
HUMIDITY
=75°F. EF. TEMP.

Fig. 9.
64°F. Wet
Bulb or 28.9?
B.t.u. per
Pound

82° F. DRY-BULB
AND
50%RELATIVE
HUMIDITY
=75° F. EF. TEMP.
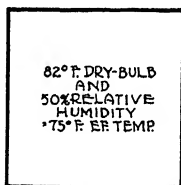
Fig. 10.   68°F. Wet
Bulb or 31.92 B.t.u.
per Pound

78° F. DRY-BULB
AND
70% RELATIVE
HUMIDITY
=75° F. EF. TEMP.
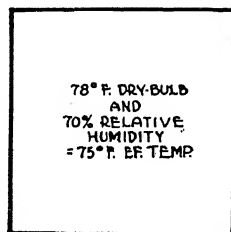
Fig. 11. 71°F. Wet Bulb or
34.33 B.t.u. per Pound

temperature line. (The effective temperature lines are slightly curved.)

From this example it can be seen that various combinations of temperature, humidity, and air movement give a composite index which is termed *effective temperature*. The effective temperature measures *effect* produced by heat and cold on a body. This, however, must not be confused with *physical comfort*.

Various explanations have been advanced as to why one feels comfortable at a higher effective temperature in the summer than in the winter. Part of the explanation is based upon the differences in diet and clothing at the different seasons. It is also probable that part of this difference may be mental attitude. In any case, this zone was plotted from results of actual tests upon subjects under specified conditions rather than from any theoretical consideration.

The Comfort Chart, Fig. 8, and also the tables of Effective Temperatures, Tables 18 and 19, are based on experimental data. That is, they represent the conditions under which the greatest per-

centage of people said they were comfortable. Because of the origin, there is some disagreement between the chart and the tables. It is suggested that the tables are more authoritative than the chart.

Table 18. Effective Temperatures Ranging from 64°F. to 69°F. for Various Dry=Bulb Temperatures and Relative Humidities for Still Air for Persons Normally Clothed and Slightly Active
(For use when heating and humidification are required)

Dry-Bulb Tempera-tures (Degrees F.)	Relative Humidities (per cent)						
	30	35	40	45	50	55	60
	Effective Temperatures (Degrees)						
67	....	....	....	....	....	64.0	64.3
68	....	....	64.0	64.2	64.5	64.8	65.1
69	64.1	64.4	64.8	65.1	65.4	65.7	66.0
70	64.8	65.1	65.4	65.8	66.2	66.5	66.8
71	65.5	65.8	66.2	66.6	67.0	67.3	67.7
72	66.2	66.5	66.9	67.3	67.7	68.1	68.5
73	67.0	67.3	67.7	68.1	68.5	68.9	....
74	67.7	68.0	68.4	68.8	....	....	....
75	68.4	68.7	....	....	....	....	....
76	69.0	....	....	....	....	....	....

Table 19. Effective Temperatures Ranging from 69°F. to 73°F. for Various Dry=Bulb Temperatures and Relative Humidities for Still Air for Persons Normally Clothed and Slightly Active
(For use when cooling or dehumidification is required)

Dry-Bulb Tempera-tures (Degrees F.)	Relative Humidities (per cent)						
	30	35	40	45	50	55	60
	Effective Temperatures (Degrees)						
73	....	....	....	....	....	....	69.3
74	....	....	....	....	69.3	69.7	70.1
75	....	....	69.1	69.5	70.0	70.5	71.0
76	69.0	69.4	69.9	70.5	70.8	71.3	71.8
77	69.7	70.2	70.7	71.2	71.6	72.1	72.6
78	70.4	70.9	71.4	71.9	72.4	73.0	....
79	71.1	71.6	72.2	72.6	....	....	....
80	71.8	72.4	72.9	....	....	....	....
81	72.5	....	....	....	....	....	....

Figs. 9, 10, and 11 represent three distinctly different combinations of dry-bulb temperatures and relative humidities, each one having the *effective temperature* of 75°F. Air motion is a constant factor in each case. The figures are in proportion, approximately, to the *total heat* in each case, at the given wet-bulb temperature.

The point to be stressed is that human beings are not necessarily comfortable even though the conditions to which they are subjected come within the generally accepted effective temperature range. For

instance, the conditions represented by Fig. 9, with its *low* humidity and high dry-bulb temperature, will result in a more comfortable condition than those shown in Fig. 11, where the dry-bulb temperature is 7 degrees lower. The cause of discomfort is the excessive humidity.

Let us approach this in another way. It has been stated that the total heat in air is measured by the wet-bulb thermometer. Now, if we refer to the Psychrometric Chart and plot the conditions in Fig. 9, we see that the wet-bulb temperature is 64°F., while the wet-bulb temperature under the conditions in Fig. 11 is 71°F.—an *increase* of 7 degrees in the wet-bulb temperature.

Now compare Fig. 10 with Fig. 11. There is a 4-degree dry-bulb temperature difference in favor of Fig. 11, yet the wet-bulb temperature in Fig. 10 is 3 degrees lower than in Fig. 11. This is because the relative humidity has been reduced from 70% to 50%. Or, compare Fig. 10 with Fig. 9. Here we have a 3-degree dry-bulb differential in favor of Fig. 10, yet the wet-bulb temperature of Fig. 9 is 4 degrees lower than Fig. 10. This is because the relative humidity has been reduced from 50% to 30%.

Carrying this one step further, let us assume a condition of 85°F. dry-bulb temperature and 40% relative humidity, or a wet-bulb temperature of 67°F. The effective temperature of this combination is 76°F, which is outside the comfort zone. But note that the wet-bulb temperature is actually 4 degrees lower than in Fig. 11, 1 degree lower than in Fig. 10 and 3 degrees higher than in Fig. 9. This particular combination is being maintained in many conditioned areas, giving perfect comfort, *without* the use of mechanical refrigeration.

These examples show that any variation in the relative humidity has a definite relation to the total heat in air, and to body comfort. Since the total heat in air is measured solely by the wet-bulb temperature, it follows that *comfort* is a function of the wet-bulb temperature. This statement, while it may come as a new concept to many, is well substantiated.

No one set of conditions or combination of temperature and relative humidity can be recommended as "ideal," nor can any specific system be considered the "best" system, unless it is based on a comprehensive study of local conditions leading to the desired results within economic limits. Any number of systems can be
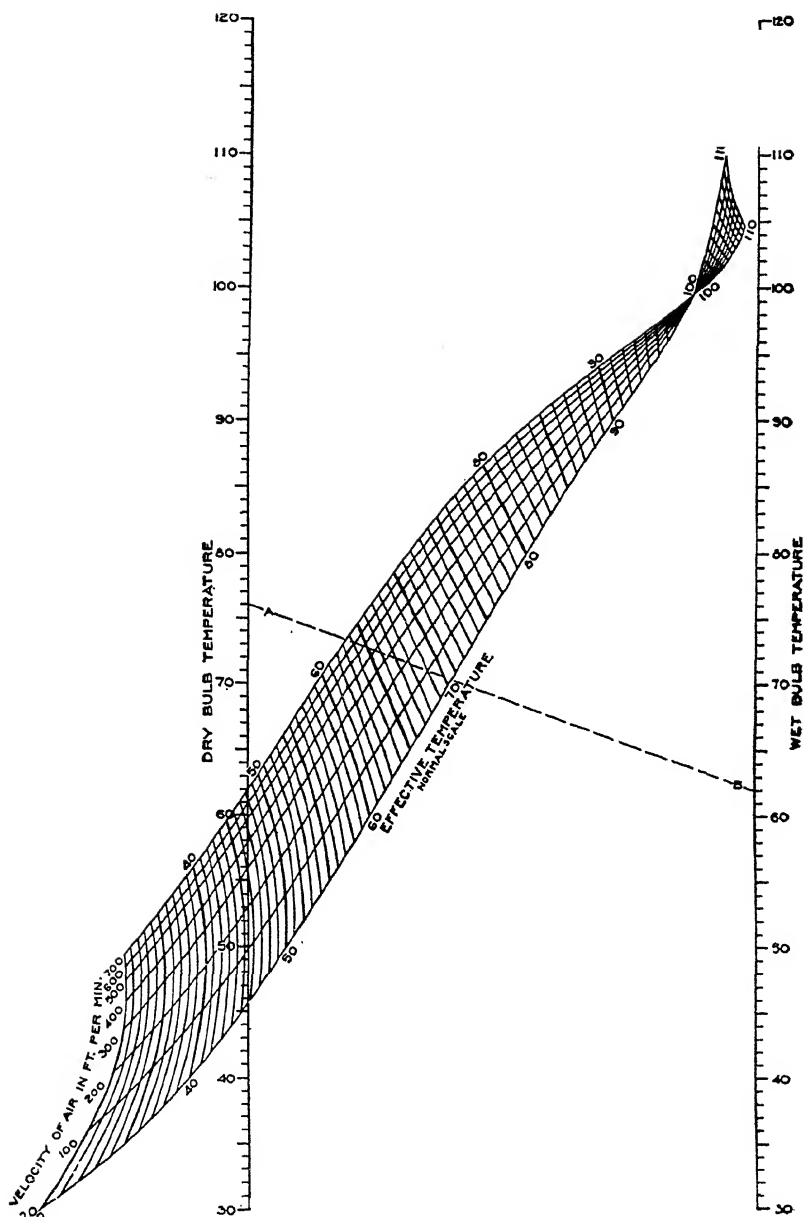
Fig. 12. Thermometric or Effective Temperature Chart Showing Normal Scale of Effective Temperature.
Applicable to Inhabitants of the United States under Following Conditions:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.* warm air, direct steam or hot water radiators, plenum systems.

Courtesy of American Society of Heating and Ventilating Engineers Guide, 1937

designed to produce a cooling effect, but only one will be most flexible and economical to own and operate.

The American Society of Heating and Ventilating Engineers carried out a series of tests in the psychrometric rooms of their Research Laboratory in Pittsburgh, to determine the equivalent conditions encountered in general air-conditioning work. Fig. 12 shows the results in a Thermometric Chart. The equivalent conditions or effective temperature lines are shown by the short cross-lines. The use of Fig. 12 can be shown by a typical example.

Example. Given dry-bulb and wet-bulb temperatures of 76°F. and 62°F., respectively, and an air velocity of 100 f.p.m., determine: (1) effective temperature of the condition; (2) effective temperature with still air; (3) cooling produced by the movement of the air; (4) velocity necessary to reduce the condition to 66°F. effective temperature.

Solution. (1) In Fig. 12 draw line *AB* through given dry- and wet-bulb temperatures. Its intersection with the 100-foot velocity curve gives 69°F. for the effective temperature of the condition. (2) Follow line *AB* to the right to its intersection with the 20 f.p.m. velocity line, and read 70.4°F. for the effective temperature for this velocity (or so-called "still air"). (3) The cooling produced by the movement of the air is $70.4 - 69 = 1.4$°F. effective temperature. (4) Follow line *AB* to the left until it crosses the 66°F. effective temperature line. Interpolate velocity value of 340 f.p.m. to which the movement of the air must be increased for maximum comfort.

Determining Sensible and Latent Heat. This can be explained by the examples which follow:

*Example 1. How much sensible heat, how much latent heat, and how much water vapor will be added per hour to the atmosphere of an assembly hall by an audience of 1000 adults, when the dry- and wet-bulb temperatures are 75°F. and 63.5°F.

Solution. Study curve *D* in Fig. 13. Note that the sensible heat loss per person for a dry-bulb temperature of 75°F. and still air is 265 B.t.u. per hour. In Fig. 14 we see that the latent heat loss per person for a dry-bulb temperature of 75°F. is 134 B.t.u. per hour and that the moisture added is 905 grains per hour. Therefore

Sensible heat $= 1000 \times 265 = 265,000$ B.t.u.
Latent heat $= 1000 \times 134 = 134,000$ B.t.u.
Water vapor added $= 1000 \times 905 = 905,000$ grains or,
$\qquad 905,000 \div 7000\dagger = 129$ pounds.

The sensible and latent heat added to the air also may be determined as follows: In Fig. 12 (Thermometric Chart) we see that the effective temperature for dry- and wet-bulb temperatures of 75°F. and 63.5°F. is 70.3°F. Then, from curve *D*, Fig. 234, we see that 403 B.t.u. is the total heat added to the air by a

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*Data Courtesy of A.S.H.V.E. Guide
† Grains per pound

Fig. 14. Latent Heat and Moisture Loss from the Human Body by Evaporation, in Relation to Dry-Bulb Temperature for Still Air Conditions[a]

[a]Curve A—Men working 66,150 ft-lb per hour. Curve B—Men working 33,075 ft-lb per hour. Curve C—Men working 16,538 ft-lb per hour. Curve D—Men seated at rest. Curves A and C drawn from data at a dry-bulb temperature of 81.3°F. only and extrapolating the relation between Curves B and D which were drawn from data at many temperatures.

Fig. 13. Relation between Sensible Heat Loss from the Human Body and Dry-Bulb Temperature for Still Air[a]

Courtesy of American Society of Heating and Ventilating Engineers Guide, 1937

person for an effective temperature of 70.3°F. In Fig. 15 we find that the percentage of sensible and latent heat for the given temperature of 75°F. dry-bulb is 66.5% and 33.5%.

Then

Sensible heat $= 1000 \times .665 \times 403 = 267,995$ B.t.u.
Latent heat $= 1000 \times .335 \times 403 = 135,005$ B.t.u.

Example 2. Neglecting the gain or loss of heat to an assembly hall by transmission or infiltration, how many cubic feet of outside air, with dry- and wet-bulb temperatures of 65°F. and 59°F. respectively, 63.1° effective temperature (ET), must be supplied per hour to an assembly hall containing 1000 people to keep the inside from exceeding 75°F. dry-bulb and 65°F. wet-bulb?

Solution. Figs. 13 and 14 give 265 B.t.u. sensible heat and 905 grains of moisture as the additions per person with a dry-bulb temperature of 75°F. in the assembly hall. Therefore, 265,000 B.t.u. of the sensible heat and 905,000 grains of moisture will be added to the air in the assembly hall each hour.

Taking 0.24 as the specific heat of air, 2.4 B.t.u. per pound of air will be required to raise the dry-bulb temperature from 65°F. to 75°F. and $\dfrac{265,000}{2.4} =$ 110,400 pounds of air or $110,400 \times 13.4 = 1,479,000$ c.f.h. of air will be required. This is equivalent to $\dfrac{1,479,000}{1000 \times 60} = 24.7$ c.f.m. per person.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition is 65 grains. Therefore, the increase in moisture content will be 11 grains per pound of dry air. Hence, $\dfrac{905,000}{11.0} = 82,300$ pounds of air at the specified condition will be required. This is equivalent to $82,300 \times 13.4 = 1,103,000$ c.f.h. of air or $\dfrac{1,103,000}{1,000 \times 60} = 18.4$ c.f.m. of air per person.

The higher volume of 24.7 c.f.m. per person will be required to keep the dry-bulb temperature from rising above the 75°F. specified. The wet-bulb temperature will therefore not rise to the maximum of 65°F.
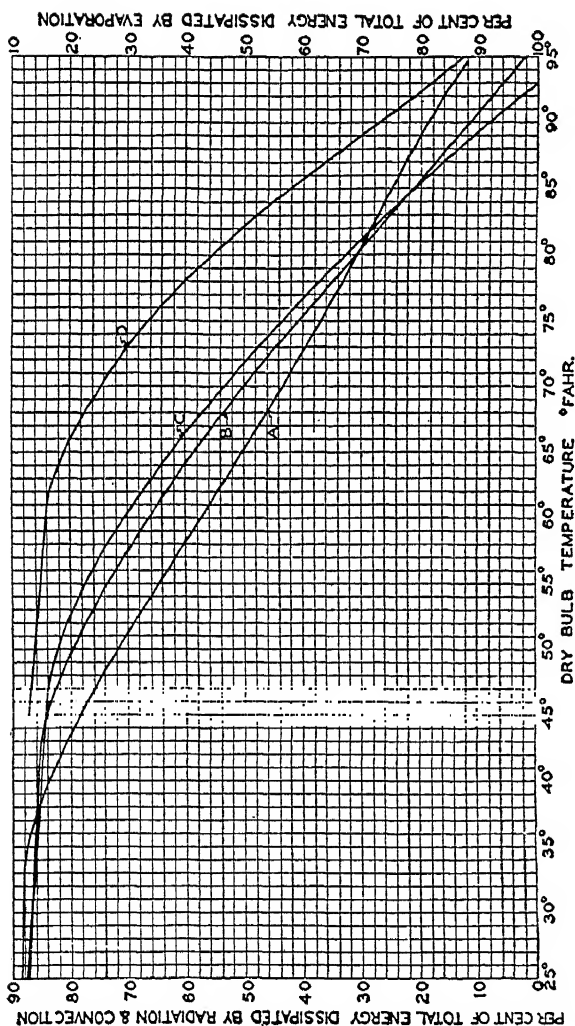
Fig. 15. Heat Loss from the Human Body by Evaporation, Radiation and Convection in Relation to Dry-Bulb Temperature for Still Air Conditions[a]

[a]Curve A—Men working 66,150 ft-lb per hour. Curve B—Men working 33,075 ft-lb per hour. Curve C—Men working 16,538 ft-lb per hour. Curve D—Men seated at rest. Curves A and C drawn from data at rest. Curves B and D which were drawn from bulb temperature of 81.3°F. only and extrapolating the relation between Curves B and D which were drawn from data at many temperatures.

Courtesy of American Society of Heating and Ventilating Engineers Guide, 1937
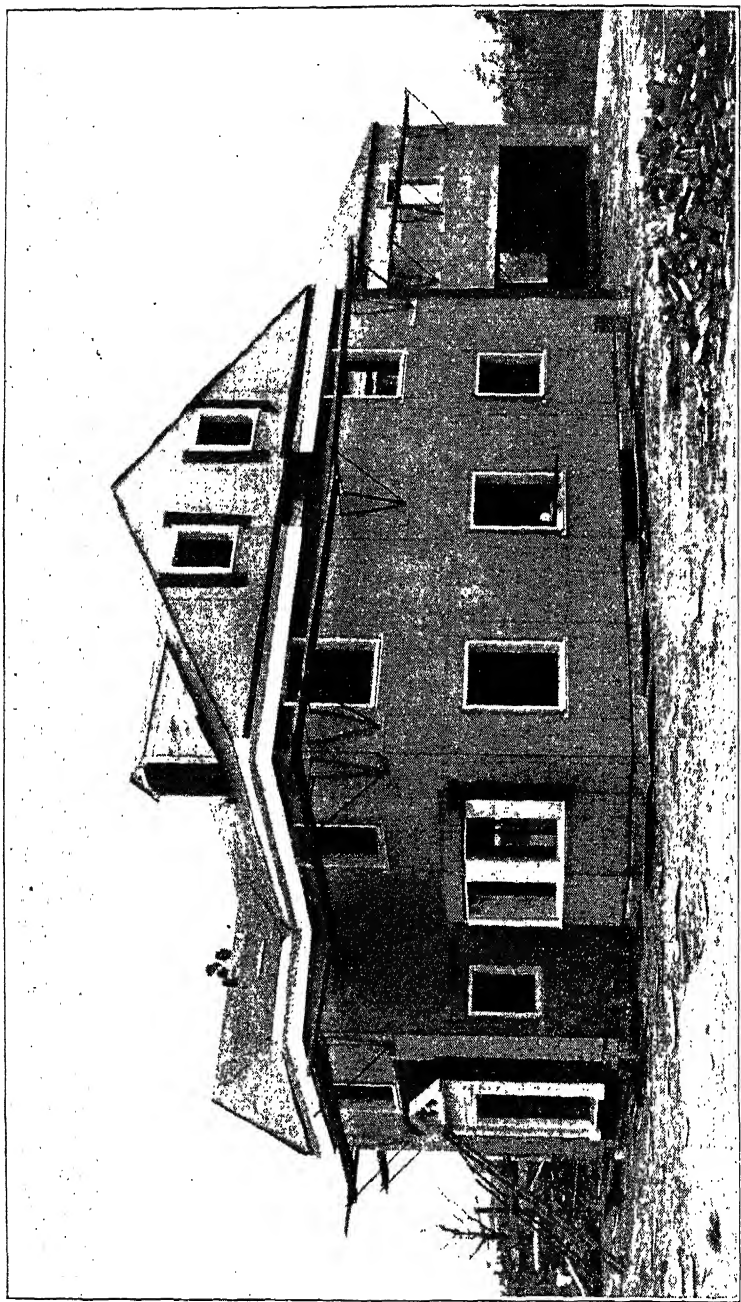
APPLICATION OF RIGID TYPE INSULATION IN PLACE OF WOOD SHEATHING

Courtesy of The Insulite Co., Minneapolis, Minn.

CHAPTER IV

INSULATION

A thorough explanation of the principles and applications of insulation (to prevent heat losses and gains in buildings), is not within the scope of this volume. Insulation is a science of its own and requires a book-length explanation to give the reader a thorough acquaintance with it and its application.

The sections on "Transmission Coefficients and Tables" and "Heat Losses and Gains," in Chapter II, present discussions of the coefficients and formulas necessary for the calculations of heat losses and gains together with several illustrative problems. In some of these problems the use of insulation is shown.

Generally speaking, insulation is a material used with, or in place of, structural building parts for the purpose of retarding or preventing heat losses and gains. For example, in the winter, insulation guards against the loss of heat through walls, roofs, floors, glass, doors, etc., and in the summer, the same insulation retards heat gain.

During the winter, a heating system functions to supply heat and to replace lost heat. Ordinary building materials such as brick, concrete, plaster, wood, glass, etc., have little ability to prevent heat transmission. Therefore, insulation materials, which possess high resistance to the transmission of heat, are used in walls, roofs, etc., either as an extra material or in place of wood sheathing, wood or metal lath, etc. During the summer, when it is desirable to keep the inside of buildings cool, the insulation keeps the outside heat from entering.

The principle of insulation is based upon its resistance to transmission of heat and upon the theory that heat tends to travel from high to low temperature. Most insulation functions on the theory of dead or still air spaces. A material containing many thousands of cell-like dead air spaces per small area tends to retard the flow of heat because dead air does not convey heat. On the other hand, brick, being very dense in structure, contains relatively few dead air spaces and conducts heat readily. The high-to-low theory of heat

travel means that in the winter the high temperatures from the inside of buildings, unless halted by insulation, travel through structural parts to the low outside temperatures. In the summer the high outside temperatures travel to the lower inside temperatures.

Some insulations function on the theory of reflection and air spaces between thin layers of the material itself. Bright metallic materials, thin as paper, possess the ability to reflect heat and thus serve as insulation. If layers of such insulating materials are spaced some distance apart, the spaces also serve to insulate.

The type of insulation used in roofs should be selected with the heat storage theory in mind. The thinner the insulation material, the less bulk; and the less bulk, the less heat storage. Most roofs are subject to direct and almost perpendicular rays from the sun, with the result that roof members or parts are exposed to heat many degrees higher than air temperature. This causes storage of heat in the roof bulk. This stored heat, which is released after sundown, enters the rooms nearest the roof. Bulky insulations are not advisable in roofs because they, too, store heat.

If residences or industrial buildings require high relative humidities, some insulation must be used to prevent condensation on walls and under sides of roofs—especially concrete roofs.

Ducts, steam pipes, water pipes, and boilers may all be insulated to safeguard them against heat loss, heat gain, freezing, etc.

Insulation brings about fuel savings in winter and, where summer conditioning is carried on, it brings about economies in electricity, etc. Insulation should, in all cases, be economical as well as comfortable. It is possible to insulate beyond the point where the cost of insulation is offset by fuel savings or is too great compared to the comforts obtained. Therefore, the application of insulation should be studied with care before any decision is made.

CHAPTER V

BOILERS

Tubular Boilers. Tubular boilers are used largely for heating and are adaptable to all classes of buildings, except dwellings and special cases mentioned later for which sectional boilers are preferable.

Boiler horsepower has been defined as the evaporation of $34\frac{1}{2}$ pounds of water from and at a temperature of 212°F. In this process 33,523 B.t.u. are absorbed, which are again given out when the steam is condensed in the radiators. Hence, to find the boiler horsepower required for warming any given building, we have only to compute the heat loss per hour, by the methods already given, together with piping losses and divide the result by 33,523. (It is good policy to divide by the number 33,000, which gives a slightly larger boiler and is on the side of safety.)

The commercial horsepower of a well-designed boiler is based upon its heating surface; and for the best economy in heating work, it should be proportioned to have about 1 square foot of heating surface for each 2 pounds of water to be evaporated from and at 212°F. This gives $34.5 \div 2 = 17.2$ square feet of heating surface per horsepower, which is generally taken as 15 in practice. Makers of tubular boilers commonly rate them on a basis of 12 square feet of heating surface per horsepower. This is a safe figure under the conditions of power work where skilled firemen are employed and where care is taken to keep the heating surfaces free from soot and ashes. For heating plants, however, it is better to rate the boilers upon 15 square feet per horsepower, as stated above.

There is difference of opinion concerning the proper method of computing the heating surface of tubular boilers. In general, all surface is taken which is exposed to the hot gases on one side and to the water on the other. A safe rule is to take $\frac{1}{2}$ the area of the shell, $\frac{2}{3}$ of the rear head less the tube area, and the interior surface of all the tubes.

The required amount of grate area and the proper ratio of heating surface to grate area vary a good deal, depending on the character

of the fuel and on the chimney draft. By assuming the probable
rates of combustion and evaporation, we may compute the required
grate area for any boiler from the formula

$$S = \frac{hp. \times 34.5}{E \times C} \tag{14}$$

where

S = total grate area in square feet

E = pounds of water evaporated per pound of coal

C = pounds of coal burned per square foot of grate per hour

Table 20 gives the approximate grate area per horsepower for
different rates of evaporation and combustion as computed by the
above formula.

**Table 20.   Grate Area per Horsepower for Different Rates of
Evaporation and Combustion**

Pounds of Steam per Pound of Coal	Pounds of Coal Burned per Square Foot of Grate per Hour		
	8 Lbs.	10 Lbs.	12 Lbs.
	Square Feet of Grate Surface per Horsepower		
10	.43	.35	.28
9	.48	.38	.32
8	.54	.43	.36
7	.62	.49	.41
6	.72	.58	.48

For example, with an evaporation of 8 pounds of steam per pound of coal
and a combustion of 10 pounds of coal per square foot of grate, .43 of a square
foot of grate surface per horsepower would be called for.

The ratio of heating to grate surface in this type of boiler ranges
from 30 to 40, and therefore allows under ordinary conditions a com-
bustion of from 8 to 10 pounds of coal per square foot of grate. This
is easily obtained with a good chimney draft and careful firing.
Usually, the larger the boiler, the more important the plant and the
greater the care bestowed upon it, so we may generally count on a
higher rate of combustion and a greater efficiency as the size of the
boiler increases. Table 21 will be found very useful in determining
the size of boiler required under different conditions. The grate area
is computed for an evaporation of 8 pounds of water per pound of
coal, which corresponds to an efficiency of about 60%, and is about
the average obtained in practice for heating boilers.

The areas of uptake and smoke-pipe are figured on a basis of

1 square foot to 7 square feet of grate surface, and the results given in round numbers. In the smaller sizes the relative size of smoke-pipe is greater. The rate of combustion runs from 6 pounds in the smaller sizes to 11½ in the larger. Boilers of the proportions given in the table correspond well with those used in actual practice and may be relied upon to give good results under all ordinary conditions.

Table 21. Size of Boiler Required Under Different Conditions

Diameter of Shell (In Inches)	Number of Tubes	Diameter of Tubes (In Inches)	Length of Tubes (In Feet)	Horse-Power	Size of Grate (In Inches)	Size of Uptake (In Inches)	Size of Smoke-pipe (In Sq. In.)
30	28	2½	6	8.5	24×36	10×14	140
			7	9.9	24×36	10×14	140
			8	11.2	24×36	10×14	140
			9	12.6	24×42	10×14	140
			10	14.0	24×42	10×14	140
36	34	2½	8	13.6	30×36	10×16	160
			9	15.3	30×42	10×18	180
			10	16.9	30×42	10×18	180
			11	18.6	30×48	10×20	200
			12	20.9	30×48	10×20	200
42	34	3	9	18.5	36×42	10×20	200
			10	20.5	36×42	10×20	200
			11	22.5	36×48	10×25	250
			12	24.5	36×48	10×25	250
			13	26.5	36×48	10×28	280
			14	28.5	36×54	10×28	280
48	44	3	10	30.4	42×48	10×28	280
			11	33.2	42×48	10×28	280
			12	35.7	42×54	10×32	320
			13	38.3	42×54	10×32	320
			14	40.8	42×60	10×36	360
			15	43.4	42×60	10×36	360
			16	45.9	42×60	10×36	360
54	54	3	11	34.6	48×54	10×38	380
			12	37.7	48×54	10×38	380
			13	40.8	48×54	10×38	380
			14	43.9	48×54	10×38	380
			15	47.0	48×60	10×40	400
			16	50.1	48×60	10×40	400
	46	3½	17	53.0	48×60	10×40	400
60	72	3	12	48.4	54×60	12×40	460
			13	52.4	54×60	12×40	460
			14	56.4	54×60	12×40	460
			15	60.4	54×66	12×42	500
			16	64.4	54×66	12×42	500
	64	3½	17	71.4	54×72	12×48	550
			18	75.6	54×72	12×48	550
66	90	3	14	70.1	60×66	12×48	500
			15	75.0	60×72	12×52	620
			16	80.0	60×72	12×52	620
	78	3½	17	86.0	60×78	12×56	670
			18	91.1	60×78	12×56	670
			19	96.2	60×78	12×56	670
	62	4	20	93.1	60×78	12×56	670
72	114	3	14	87.4	66×72	12×56	670
			15	93.6	66×72	12×56	670
			16	99.7	66×78	12×62	740
	98	3½	17	106.4	66×78	12×62	740
			18	112.6	66×84	12×66	790
			19	118.8	66×84	12×66	790
	72	4	20	107.3	66×84	12×66	790

Water-tube boilers are often used for heating purposes, but more especially in connection with power plants. The method of computing the required horsepower is the same as for tubular boilers.

Sectional Boilers. A common type of cast-iron boiler is shown in Fig. 16. It is made of circular sections which are connected to-
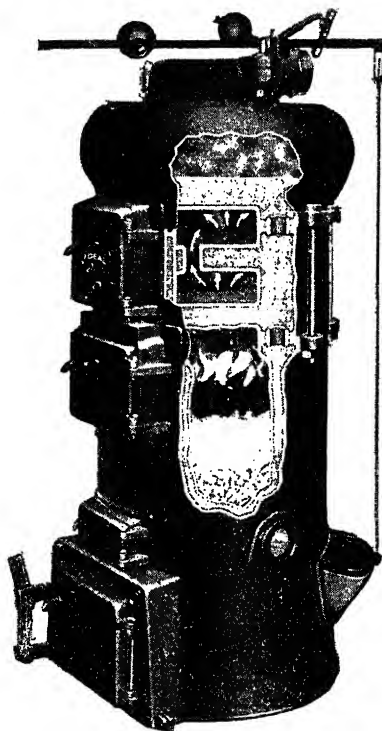
Fig. 16. Round Cast-Iron Sectional Boiler

gether at the top and bottom by nipples. This boiler is known as a round boiler and is used for heating small residences. Two steam outlet connections are on top of the upper section. The boiler return connections are at about the level of the grate and one is located on each side of the firing door. The gases from the fire pass upward and escape through smoke connection attached to the top section.

Another type of sectional cast-iron boiler is shown in Fig. 17. This boiler is a horizontal type and the sections are fastened together by nipples located at the top and bottom of the sections.

The steam outlet connections at the top of the boiler should be piped into a common steam header. Each boiler section having a steam outlet tapping at the top has two return tappings at the bottom, one located on each side of the boiler. All of these return tappings should be piped through a yoke connection around the
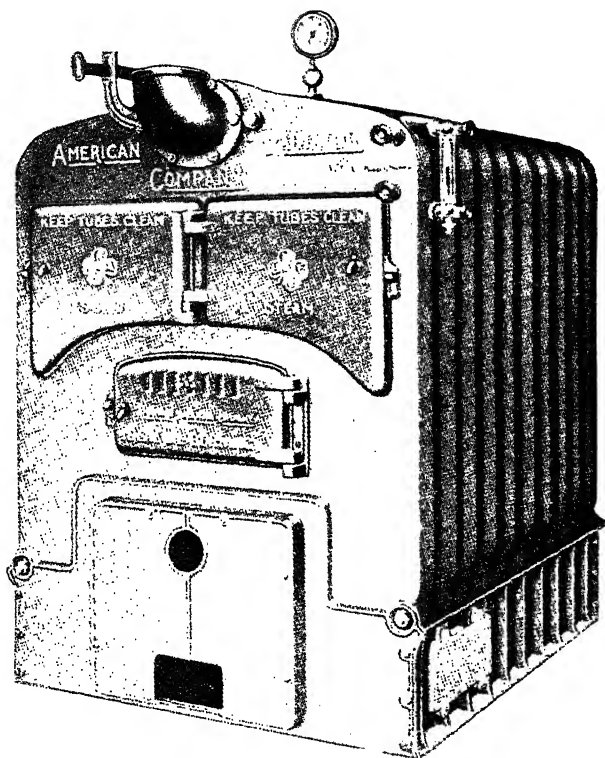
Fig. 17. Cast-Iron Sectional Boiler

boiler to the return line from the heating system. The flue gases from the fire pass back and forth through the openings in the sections and escape to the chimney through a smoke connection located at the back of the boiler.

These boilers when used for steam are fitted with a pressure gauge, gauge cocks, water glass, and safety valve. A low pressure damper regulator may be provided for opening and closing the draft doors. Steam pressures ranging from 1 to 5 pounds per square inch

are usually carried, depending upon the type of heating system and the weather conditions. Boilers should be well insulated with some good plastic magnesia or asbestos covering. Insulation should also be placed on all basement mains and piping carrying steam to the radiators and condensed steam from the radiators.

Certain makers of cast-iron horizontal sectional boilers for steam or hot-water heating provide the boiler with drums at the top and bottom connected by nipples to each section. Such an arrangement may give dryer steam and hold a steadier water line. Cast-iron sectional boilers have more or less restricted openings between the sections and when forced beyond their normal capacity are likely to have the openings choked. This may result in having water carried with the steam or, in the case of excessive forcing, in having the water emptied out of the boiler into the steam mains. When these boilers are used for hot water this is not true. The boilers when used for steam should have ample steam space, good-sized connections between sections, and ample openings in the steam nozzles or outlets.

Cast-iron boilers are suitable for steam or hot-water heating in dwellings, small school houses, churches, etc., where low pressures are used. They can be increased in size by adding sections up to a certain limit. Beyond the limit of about 10 sections, they become less and less efficient, so their size and power are limited.

Boiler sizes are best computed on the basis of square feet of grate area required and not on the boiler rating in square feet of radiation. The amount of radiation that a boiler will supply is dependent upon the type of radiation, the operating conditions, and the line losses between the boiler and the radiators.

Good house heating boilers attached to a good chimney may be expected to burn about 8 pounds of coal per square foot of grate area per hour with an efficiency of about 55%.

If coal having a heat value of 12,500 B.t.u. per pound is used with an efficiency of 55%, the heat utilized per pound of fuel is

$$12,500 \times 0.55 = 6,875 \text{ B.t.u.}$$

It is customary to add to the heat loss from the building a certain percentage to cover the losses in the piping, etc., between the boiler and the radiators in the rooms to be heated. This percentage may range from 20 for large plants up to 50 for very small installations.

For residences use 25%. The calculation of the required size of a boiler is illustrated by the following example:

Example. The heat loss from a building is 400,000 B.t.u. per hour. What size cast-iron boiler is required?

Solution: Heat losses plus
25% piping losses $= 400,000 \times 1.25 = 500,000$ B.t.u. per hour
Pounds of coal per hour $= 500,000 \div 6875 = 73$ (approx.)
Grate area based on 8 pounds of coal
per square foot $\qquad = 73 \div 8 = 9.12$ square feet

If the coal has a heat value lower than 12,500 B.t.u. per pound or if the chimney is poor, then lower values must be used for the heat available per pound of fuel or for the combustion rate per hour.

Steel Boilers. A representative type of a steel heating boiler is shown in Fig. 18. Boilers of this type are suitable for very large residences, schools, apartment buildings, and office buildings. Steel heating boilers are usually built for working pressures up to 15 pounds per square inch and are capable of greater capacity than cast-iron sectional boilers.

The boiler illustrated is a brick set, down-draft, smokeless type. It may be used for either steam or hot-water heating. The path of the flue gas travel is clearly shown in the illustration. The fuel is fired on the upper water-tube grate, and the products of combustion pass downward through the upper grate and over the incandescent fuel bed on the lower grate. This tends to produce smokeless combustion. No fuel is fired directly on the lower grate as a sufficient amount of burning fuel falls from the upper to the lower grate.

The necessary size of such a boiler may be calculated in the same manner as was shown for a cast-iron boiler. The boiler should be selected on the basis of upper grate area. When good chimney draft is available, 10 pounds of coal may be fired per hour per square foot of upper grate area.

It should be noted that for steel boilers the rate of combustion is 10 instead of 8, as previously used for furnaces and cast-iron boilers.

PRACTICE PROBLEMS

1. The heat loss of a building is 240,000 B.t.u. per hour. The percentage of heat loss is 32%, coal having a heat value of 8,000 B.t.u. per pound is used, and the efficiency is 55%. What will be the required grate area?

2. The heat loss of a building is 168,000 B.t.u. per hour, and the chimney draft is such that not over three pounds of coal per hour can be burned per square
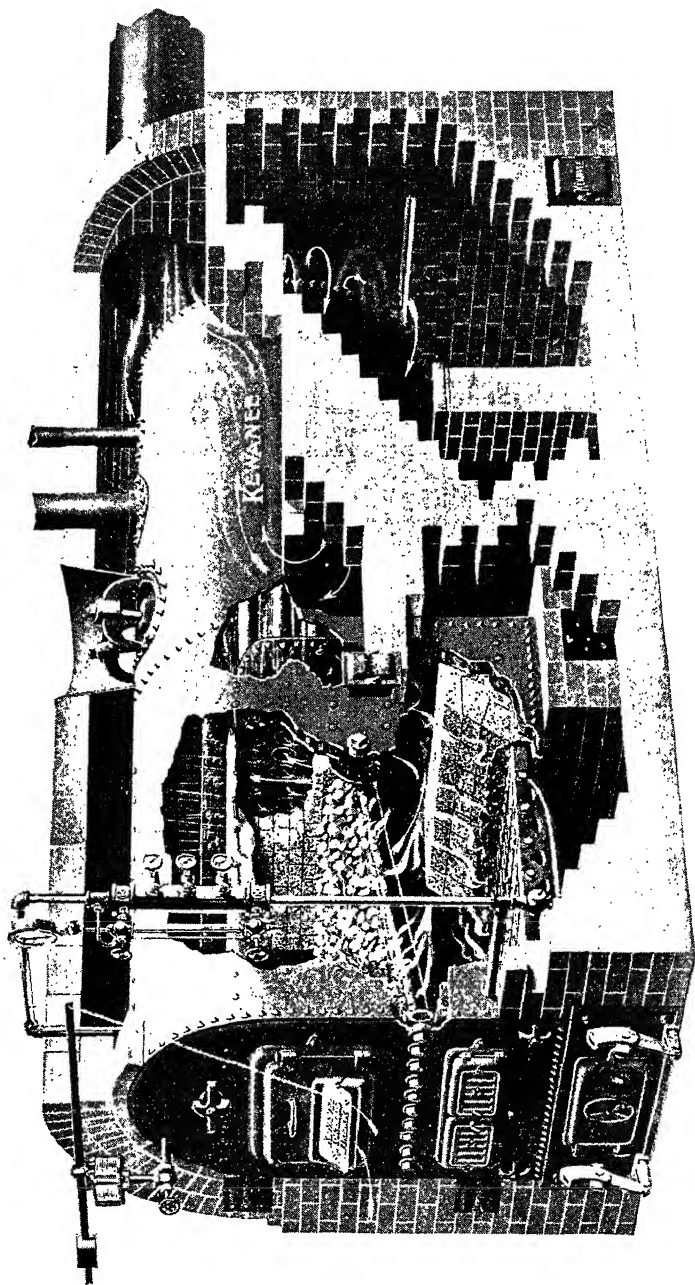
Fig. 18. Down-Draft, Brick Set, Steel Heating Boiler

foot of grate. The pipe loss is 25% and the coal has a heat value of 8,000 B.t.u. per pound at 55% efficiency. What is the required grate area?

Boiler Horsepower for Ventilation. It is known that 1 B.t.u. will raise the temperature of 1 cubic foot of air 55 degrees, or it will raise 100 cubic feet $\frac{1}{100}$ of 55 degrees, or $\frac{55}{100}$ of 1 degree; therefore, to raise 100 cubic feet 1 degree, it will take $1 \div \frac{55}{100}$, or $\frac{100}{55}$ B.t.u.; and to raise 100 cubic feet through 100 degrees, it will take $\frac{100}{55} \times 100$ B.t.u. In other words, the B.t.u. required to raise any given volume of air through any number of degrees in temperature, is equal to

$$\frac{\text{Volume of air in cubic feet} \times \text{degrees raised}}{55}$$

Example. How many B.t.u. are required to raise 100,000 cubic feet of air 70 degrees?

Solution. $\qquad \frac{100,000 \times 70}{55} = 127,272+$

To compute the horsepower required for the ventilation of a building, we multiply the total air-supply, in cubic feet per hour, by the number of degrees through which it is to be raised, and divide the result by 55. This gives the B.t.u. per hour, which divided by 33,000, will give the horsepower required. In using this rule, always take the air-supply in cubic feet per hour.

PRACTICE PROBLEMS

1. The heat loss from a building is 1,650,000 B.t.u. per hour. There is to be an air-supply of 1,500,000 cubic feet per hour, raised through 70 degrees. What is the total boiler horsepower required?

2. A high school has 10 classrooms, each occupied by 50 pupils. Air is to be delivered to the rooms at a temperature of 70°F. What will be the total horsepower required to heat and ventilate the building when it is −10°F. if the heat loss by conduction, etc., is 1,320,000 B.t.u. per hour and the pipe losses 20%?

Jacketed Boilers. The boilers thus far described are of types which have been in use for many years. They have been discussed here to aid the reader in handling jobs involving them. From this point on, modern boilers will be discussed.

Sectional Boilers. Fig. 19 shows a cutaway view of a sectional boiler for coal burning. In this boiler the efficient transfer of heat from fuel to the heating medium is the primary consideration. This has been accomplished by controlling the heat level. Water rising up each leg is turned across the top of the combustion chamber by baffles. Here it rises through a passage formed by the inner sides of the first pass flues to the upper water travel where the stream is divided and flows outwardly to each side of the boiler. It then rises

to the dome and reaches the water line at points most remote from the outlet, resulting in less turbulent water line and in drier steam. The boiler is jacketed, as in Fig. 20, and makes a splendid appearance in the modern basement or other heating space.
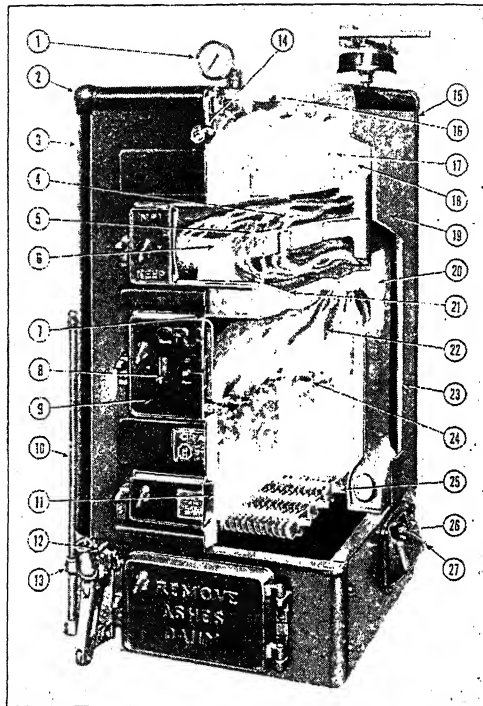
Fig. 19. Sectional Boiler for Coal Burning and for Hot Water, Steam, Vacuum, and Vapor Heating
Courtesy of Crane Co.

The following explanations apply to Fig. 19 and the corresponding numbers shown there.

(1) Boilers are shipped complete, with try-cocks, pop valves, steam gauges, combination altitude gauges and thermometers, and regulators.

(2) Four screws at top corners are tightened to pull jacket up for a perfect fit.

(3) Marring of jacket is eliminated because jacket is put on after boiler is installed and ready to operate.

(4) Broad flat gas flues have less tendency to collect soot. Scraper is inserted easily for cleaning out soot.

(5) Elongated two-pass gas flues.

(6) Here soot raked forward drops into first pass, then is pushed back to fall into combustion chamber.

(7) All doors and dampers have flat surfaces which permit them to shut tight.

(8) Fire door has secondary air distributor.
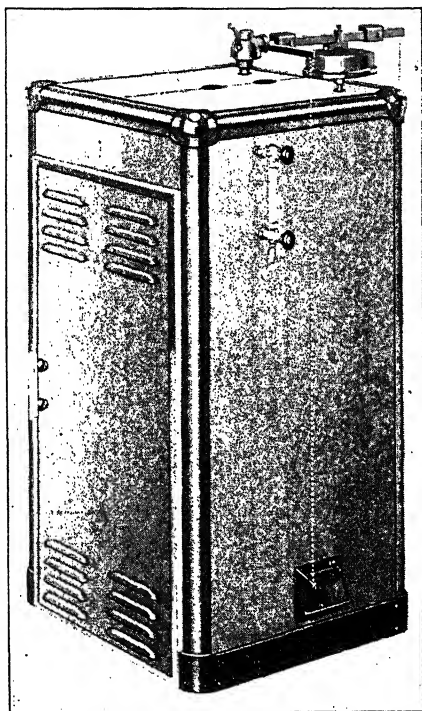
Fig. 20. Jacketed Sectional Coal Burning Boiler
Courtesy of Crane Co.

(9) Heat-resisting black paint is used for base, front, doors, and all exposed parts.

(10) Steel shaker handle.

(11) Boiler doors are equipped with baffle linings, flue doors with curved baffles for easy turning of gas flow.

(12) Grate shaker bar locking device locks with a tap of the foot. It has three positions, one for shaking, one for dumping, and one for locking in place.

(13) Socket on left side of boiler holds shaker handle when not in use. Prevents misplacing it.

(14) Surface blow-off tapping is provided.

(15) Smoke damper with positive indicator on two-way smoke hood has a locking device which holds it in position.

(16) Threaded openings.

(17) Push-nipple openings.

(18) Boiling water reaches water line at points most remote from outlet, giving less turbulent water line and drier steam.

(19) Two-tone green jacket.

(20) Patented baffles direct water across top of combustion chamber, giving controlled and longer water travel and therefore faster water travel.

(21) Flat surfaces of all sections are designed to give a tight, true fit.
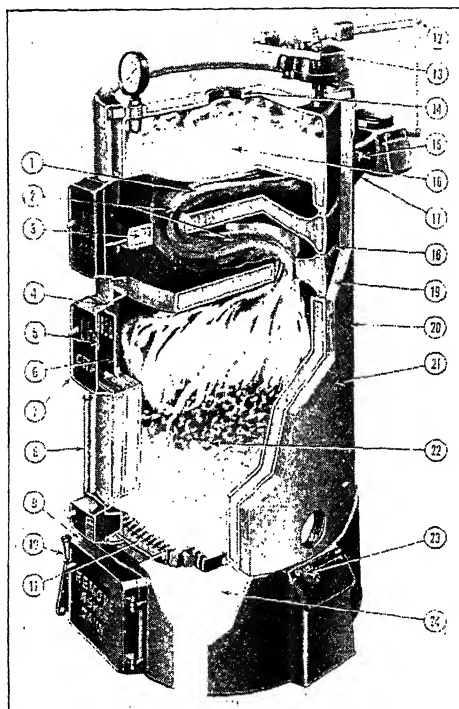
Fig. 21.   Round Boiler for Coal Burning and for Hot
Water, Steam, and Vapor Heating
Courtesy of Crane Co.

(22) The back wall, which is exposed to hottest gases, is corrugated to eliminate strains.

(23) Sides and top insulated.

(24) Grate area, combustion chamber, and heating surfaces.

(25) Non-jumping sockets for grate ends prevent grates being thrown out of place when clinkers catch between the teeth.   Grates are sloping and are designed to burn all sizes of coal, including buckwheat or pea coal.

(26) Brass pivots at bearings minimize friction and prevent corrosion that would cause damper to stick.   Insures sensitive response to control.

(27) Malleable iron damper door levels are attached to give good leverage and a sensitive, accurate, easy control by automatic regulator. (Applies to coal burning boilers only.)

Round Boilers. Fig. 21 shows a cutaway view of a round boiler
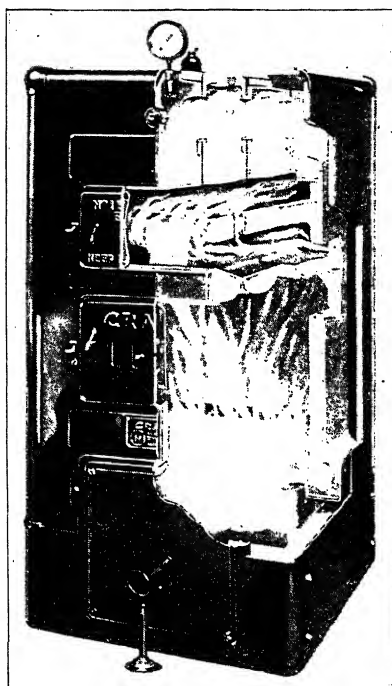
Fig. 22. Sectional Boiler Converted to Oil Burning Requirements. See Appendix for Method of Selection
Courtesy of Crane Co.

for coal burning. The following explanations apply to Fig. 21 and the corresponding numbers shown in the figure.

(1) Depression in top of gas flue at dome maintains velocity of gas as it goes to the smoke hood. This same depression gives greater water depth at dome.

(2) Arrangement of gas flues gives back and forth gas travel and consequently long fire travel.

(3) Water passages arranged for back and forth water travel increase velocity of water and afford maximum contact with heating surfaces.

(4) All doors and dampers have flat surfaces which permit them to shut tight, without air or dust leaks. Corresponding parts are interchangeable.

(5) Fire door has secondary air distributor.

(6) All boiler doors have baffle linings.

(7) Heat-resisting black paint is used for all exposed parts.

(8) Steel shaker handle.

(9) Socket at side holds the shaker handle to keep it from getting lost.

(10) Grate shaker bar locking device locks with a tap of the foot. It has three positions, one for shaking, one for dumping, and one for locking in place.

(11) Sloping grates make shaking and dumping easier.

(12) Regulator-controlled damper arms have a balanced leverage to give easy, non-sticking operation.

(13) Try-cocks, pop valves, steam gauges, combination altitude gauges and thermometers.

(14) Threaded openings for easy installation.

(15) Smoke damper with positive indicator on two-way smoke hood has a locking device which holds it in place when adjusted.

(16) Extra large dome provides ample water capacity for steam generation. (Applies to steam boilers only.)

(17) Smoke hood is at back, permitting installation with less headroom.

(18) Push-nipple openings are uniformly accurate.

(19) Insulation under jacket.

(20) Two-tone green jacket.

(21) Marring of jacket is eliminated because jacket is put on after boiler is installed and ready to operate. No unsightly cracks are left in jacket joints.

(22) Combustion chamber is extra deep, giving large fuel capacity.

(23) Brass pivots at bearings minimize friction and insure sensitive response to control. (24) Ash pit.

Oil=Burning Boilers. Fig. 22 shows a sectional boiler converted to oil burning requirements.

How to Select Correct Size of Boiler. *Tables 22 to 26 are used for rating boilers like those shown in Figs. 19 to 22. The tables use the accepted practice of expressing all boiler outputs in B.t.u. and in square feet of direct cast-iron radiation—based on the standard heat emission of 240 B.t.u. for steam, and 150 B.t.u. for hot water, per square foot of radiation surface per hour.

Hard Coal Burning. Hard coal burning outputs are figured on hard coal of commercial size having a 12,500 B.t.u. heat value. To determine the correct size hard coal boiler, choose the rating table showing (at the top of one column) the total equivalent direct radiation load desired. For a steam boiler, refer to the numbers in italics, and for a hot water boiler refer to those in bold Roman type. Select the next higher figure above the actual equivalent direct radiation load. Follow down the column to the heavy vertical parallel lines. At the left of the chart the recommended boiler is indicated.

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*Complete tables can be secured from manufacturers.

Table 22. Crane Sectional Boiler—Performance Data

Number of Boiler	Output in Equivalent Direct Radiation Sq. Ft.	100 / 160 / 24	175 / 280 / 42	250 / 400 / 60	325 / 520 / 78	400 / 640 / 96	475 / 760 / 114	550 / 880 / 132	625 / 1000 / 150	700 / 1120 / 168	775 / 1240 / 186	850 / 1360 / 204	925 / 1480 / 222	1000 / 1600 / 240	1075 / 1720 / 258	1150 / 1840 / 276	1225 / 1960 / 294	1300 / 2080 / 312	Fuel Cap. / Fuel Avail.	Stack Size and Height
	Steam (240 B.t.u.) / Water (150 B.t.u.) / B.t.u. Per Hour (1000's)																			
1-4-S / 1-4-W Grate Area 1.40 Sq. Ft.	Firing Period—Hours	33.9	19.1	13.3	10.1	8.2													125# Cap. / 95# Avail.	8x8 Ins. / 30 ft. High
	Boiler Efficiency—%	68.6	67.7	67.2	66.5	66.4														
	Stack Gas Temp.—°F	300	375	450	525	600														
	Combustion Rate	2.00	3.55	5.10	6.70	8.25														
	Draft—Ins. of Water	.030	.034	.045	.062	.085														
1-5-S / 1-5-W Grate Area 1.90 Sq. Ft.	Firing Period—Hours	26.5	18.5	14.3	11.6			8.3											170# Cap. / 130# Avail.	8x8 Ins. / 30 ft. High
	Boiler Efficiency—%	68.6	68.5	68.4	68.2			67.7												
	Stack Gas Temp.—°F	320	390	455	525			660												
	Combustion Rate	2.58	3.69	4.80	5.92			8.20												
	Draft—Ins. of Water	.030	.037	.048	.065			.110												
1-6-S / 1-6-W Grate Area 2.40 Sq. Ft.	Firing Period—Hours			24.6	18.8	15.1	12.6	10.7	9.4	8.4	7.5								215# Cap. / 165# Avail.	8x8 Ins. / 30 ft. High
	Boiler Efficiency—%			71.4	71.0	71.0	69.6	69.6	68.5	68.3	67.8									
	Stack Gas Temp.—°F			345	395	450	505	560	615	670	720									
	Combustion Rate			2.80	3.68	4.55	5.46	6.40	7.30	8.20	9.15									
	Draft—Ins. of Water			.027	.033	.046	.065	.084	.105	.130	.160									
1-7-S / 1-7-W Grate Area 2.90 Sq. Ft.	Firing Period—Hours				23.1	18.7	15.7	13.3	11.6	10.3	9.3	8.4	7.7						200# Cap. / 200# Avail.	8x12 Ins. / 35 ft. High
	Boiler Efficiency—%				72.2	71.6	71.1	70.0	69.8	69.2	68.9	68.7	68.4							
	Stack Gas Temp.—°F				330	385	440	490	540	595	650	700	755							
	Combustion Rate				2.98	3.70	4.42	5.20	5.93	6.70	7.45	8.20	8.95							
	Draft—Ins. of Water				.027	.034	.045	.060	.080	.100	.125	.150	.170							
1-8-S / 1-8-W Grate Area 3.40 Sq. Ft.	Firing Period—Hours					22.3	18.8	16.0	13.4	12.3	11.1	10.1	9.2	8.4	7.8	7.3			305# Cap. / 235# Avail.	8x12 Ins. / 35 ft. High
	Boiler Efficiency—%					72.9	72.5	72.1	71.3	71.0	70.6	70.0	69.6	68.6	68.0	68.3				
	Stack Gas Temp.—°F					325	380	425	480	525	570	625	670	725	780	825				
	Combustion Rate					3.10	3.70	4.31	4.95	5.60	6.23	6.85	7.50	8.20	8.85	9.5				
	Draft—Ins. of Water					.028	.036	.048	.062	.078	.095	.115	.138	.160	.180	.200				
1-9-S / 1-9-W Grate Area 3.90 Sq. Ft.	Firing Period—Hours						22.0	18.8	16.5	14.8	13.2	11.9	10.9	10.0	9.3	8.8	8.1	7.6	350# Cap. / 270# Avail.	8x12 Ins. / 35 ft. High
	Boiler Efficiency—%						74.2	73.3	73.0	73.0	72.5	72.1	71.7	71.3	71.0	70.8	70.5	70.3		
	Stack Gas Temp.—°F						305	345	405	445	500	555	605	645	705	750	795	850		
	Combustion Rate						3.15	3.68	4.20	4.72	5.26	5.80	6.35	6.90	7.45	8.00	8.55	9.10		
	Draft—Ins. of Water						.020	.030	.047	.060	.073	.088	.105	.120	.145	.160	.180	.200		

To, select boiler required add actual radiator load (in sq. ft.) to piping load (in equivalent sq. ft. of radiation).
This total load is the output in equivalent direct radiation to be used. Select boiler between heavy black lines having the desired firing period and combustion rate.
The above tables are based on fuel having 12,500 B.t.u. per pound.

Table 23.   Crane Sectional Boiler—Performance Data

Output in Equivalent Direct Radiation Sq. Ft.

Output in Equivalent Direct Radiation Sq. Ft.	400	525	650	775	900	1025	1150	1275	1400	1525	1650	1775	1900	2025	2150	2275	2400	2525	2650	2775	2900
Steam (⅓ B.t.u.)	400	525	650	775	900	1025	1150	1275	1400	1525	1650	1775	1900	2025	2150	2275	2400	2525	2650	2775	2900
Water (150 B.t.u.)	640	840	1040	1240	1440	1640	1840	2040	2240	2440	2640	2840	3040	3240	3440	3640	3840	4040	4240	4440	4640
B.t.u. per Hour (1000's)	96	126	156	188	216	246	276	306	336	366	396	426	456	486	516	546	576	606	636	666	696
2-5-S / 2-5-W — Firing Period—Hours	23.4	17.5	13.9	11.7	9.9	8.7	7.8														
Boiler Efficiency—%	73.4	71.9	71.0	70.5	70.0	69.3	69.1														
Stack Gas Temp.—°F.	445	520	600	675	750	820	900														
Combustion Rate	2.95	3.95	4.95	5.95	6.95	8.00	9.00														
Draft—Ins. of Water	.017	.027	.040	.055	.073	.092	.110														
2-6-S / 2-6-W — Firing Period—Hours	22.6	18.0	14.9	12.8	11.1	9.8	8.8	7.9													
Boiler Efficiency—%	73.4	72.4	71.6	71.0	70.3	69.6	69.0	68.8													
Stack Gas Temp.—°F.	500	550	610	660	710	770	830	880													
Combustion Rate	3.05	3.83	4.62	5.40	6.22	7.05	7.85	8.70													
Draft—Ins. of Water	.023	.033	.044	.054	.073	.080	.105	.125													
2-7-S / 2-7-W — Firing Period—Hours	27.5	21.9	18.2	15.5	13.6	12.0	10.8	9.8	8.9	8.2	7.6										
Boiler Efficiency—%	73.8	72.9	72.1	71.5	70.8	70.5	70.3	70.0	69.3	69.0	68.8										
Stack Gas Temp.—°F.	450	495	550	605	655	695	750	800	855	905	950										
Combustion Rate	2.50	3.14	3.78	4.43	5.10	5.75	6.40	7.10	7.75	8.40	9.10										
Draft—Ins. of Water	.022	.027	.037	.049	.062	.075	.090	.105	.120	.140	.160										
2-8-S / 2-8-W — Firing Period—Hours	25.8	21.4	18.3	15.9	14.0	12.5	11.4	10.4	9.7	8.9	8.2	7.8									
Boiler Efficiency—%	72.8	72.1	71.7	70.6	69.8	69.8	69.5	69.3	69.0	68.7	68.5	67.7									
Stack Gas Temp.—°F.	445	490	535	575	630	680	730	775	820	870	915	965									
Combustion Rate	2.68	3.22	3.76	4.30	4.86	5.48	6.04	6.60	7.15	7.75	8.40	8.95									
Draft—Ins. of Water	.019	.029	.042	.053	.065	.077	.092	.105	.120	.135	.155	.175									
2-9-S / 2-9-W — Firing Period—Hours				25.0	21.3	18.6	16.4	14.4	13.4	12.1	11.3	10.4	9.8	9.0	8.6	8.0	7.6				
Boiler Efficiency—%				73.7	73.1	71.9	71.1	71.0	71.0	70.6	70.2	69.9	69.6	69.4	69.0	68.8	68.7				
Stack Gas Temp.—°F.				470	515	550	590	630	665	710	750	785	825	870	900	935	950				
Combustion Rate				2.75	3.22	3.70	4.18	4.65	5.13	5.70	6.27	6.82	7.40	7.95	8.15	8.62	9.10				
Draft—Ins. of Water				.028	.038	.047	.057	.069	.082	.095	.108	.122	.137	.150	.170	.185	.210				
2-10-S / 2-10-W — Firing Period—Hours					24.5	21.3	18.8	16.7	15.2	13.9	12.7	11.8	10.9	10.2	9.7	9.0	8.5	8.1	7.8		
Boiler Efficiency—%					74.4	74.0	72.4	71.7	71.3	71.3	70.9	70.5	69.9	69.7	69.5	69.2	69.0	68.7	68.6		
Stack Gas Temp.—°F.					475	515	540	575	610	650	680	720	750	775	790	820	830	860	885		
Combustion Rate					2.81	3.23	3.65	4.10	4.52	4.95	5.40	5.82	6.30	6.72	7.15	7.60	8.05	8.50	8.93		
Draft—Ins. of Water					.034	.042	.052	.063	.075	.087	.100	.114	.130	.145	.160	.175	.195	.215	.230		
2-11-S / 2-11-W — Firing Period—Hours						23.8	20.9	18.5	16.7	15.6	14.3	13.2	12.3	11.6	10.8	10.2	9.8	9.2	8.8	8.3	7.8
Boiler Efficiency—%						73.4	72.9	72.5	72.0	71.6	71.6	71.0	70.9	70.5	70.3	70.1	70.0	69.6	69.3	69.3	68.8
Stack Gas Temp.—°F.						485	510	535	570	600	630	665	695	730	765	800	825	860	885	925	960
Combustion Rate						2.90	3.28	3.65	4.04	4.42	4.80	5.20	5.60	5.98	6.35	6.72	7.10	7.50	7.90	8.30	8.75
Draft—Ins. of Water						.038	.047	.052	.068	.077	.088	.100	.114	.125	.137	.150	.170	.180	.200	.215	.230

Number of Boiler	Grate Area Sq. Ft.	Fuel Cap.	Fuel Avail.	Stack Size and Hght.
2-5-S / 2-5-W	3.55	325#	245#	8x12 Ins., 35 ft. High
2-6-S / 2-6-W	4.50	410#	310#	12x12 Ins., 35 ft. High
2-7-S / 2-7-W	5.45	495#	375#	12x12 Ins., 35 ft. High
2-8-S / 2-8-W	6.40	580#	440#	12x12 Ins., 40 ft. High
2-9-S / 2-9-W	7.35	665#	505#	12x16 Ins., 40 ft. High
2-10-S / 2-10-W	8.30	750#	570#	12x16 Ins., 45 ft. High
2-11-S / 2-11-W	9.25	835#	635#	12x16 Ins., 45 ft. High

To select boiler required add actual radiator load (in sq. ft.) to piping load (in equivalent sq. ft. of radiation). This total load is the output in equivalent direct radiation to be used. Select boiler between heavy black lines having the desired firing period and combustion rate.

The above tables are based on fuel having 12,500 B.t.u. per pound.

Table 24. 18-Inch Crane Round Boiler—Performance Data

Number of Boiler	Output in Equivalent Direct Radiation Sq. Ft.	220 352 528	240 384 576	260 416 624	280 448 672	300 480 720	320 512 768	340 544 816	360 576 864	380 608 912	400 640 900	420 672 1008	440 704 1056	460 736 1008	480 768 1152	500 800 1200	520 832 1248	560 896 1344	580 928 1392	600 960 1440	Fuel Cap. Fuel Avail.	Stack Size and Height
S-184 W-184 Grate Area 1.77 Sq. Ft.	Firing Period—Hours	25.1	22.8	21.0	19.5	18.0	16.8	15.7	14.8	14.0	13.2	12.5	11.9	11.2	10.8	10.4	10.0				195# Cap. 145# Avail.	8x8 Ins. 30 ft. High
	Boiler Efficiency—%	73.4	72.3	72.3	71.3	71.5	71.5	70.8	70.5	70.2	69.9	69.6	69.3	69.0	68.8	68.4	68.4					
	Stack Gas Temp.—°F	370	400	430	450	480	510	535	560	590	615	640	670	695	720	750	775					
	Combustion Rate	3.25	3.58	3.90	4.22	4.55	4.88	5.20	5.53	5.86	6.20	6.55	6.89	7.29	7.56	7.90	8.25					
	Draft—Ins. of Water	.027	.031	.036	.041	.046	.051	.056	.062	.068	.073	.080	.086	.094	.100	.105	.115					
S-185 W-185 Grate Area 1.77 Sq. Ft.	Firing Period—Hours	25.6	23.4	21.6	19.8	18.4	17.2	16.2	15.3	14.4	13.6	12.9	12.3	11.7	11.2	10.9	10.7	9.5			105# Cap. 145# Avail.	8x8 Ins. 35 ft. High
	Boiler Efficiency—%	74.1	74.3	74.1	73.6	73.1	73.0	72.6	72.2	71.9	71.7	71.5	71.3	71.1	71.0	70.9	70.8	70.6				
	Stack Gas Temp.—°F	305	320	340	360	375	385	410	430	445	465	480	500	515	530	550	570	600				
	Combustion Rate	3.20	3.50	3.80	4.10	4.45	4.54	5.37	5.05	5.70	6.03	6.35	6.67	7.00	7.32	7.65	7.97	8.60				
	Draft—Ins. of Water	.027	.030	.035	.039	.044	.049	.054	.059	.065	.070	.076	.082	.088	.095	.100	.110	.120				
S-186 W-186 Grate Area 1.77 Sq. Ft.	Firing Period—Hours	23.8	21.8	20.5	18.8	17.6	16.5	15.6	14.8	14.0	13.3	12.1	11.6	11.1	11.0	10.7	10.7	9.4	9.1		195# Cap. 145# Avail.	8x8 Ins. 35 ft. High
	Boiler Efficiency—%	75.4	75.2	75.0	74.7	74.6	74.5	74.3	74.2	74.1	74.0	73.8	73.7	73.6	73.4	73.3	73.1	72.7	72.3			
	Stack Gas Temp.—°F	290	300	315	330	345	360	375	390	400	420	430	445	460	473	475	473	520	535	550		
	Combustion Rate	3.45	3.75	4.04	4.35	4.65	4.95	5.25	5.54	5.85	6.16	6.46	6.78	7.09	7.42	7.64	7.97	8.35	8.68	9.00		
	Draft—Ins. of Water	.030	.034	.038	.043	.047	.052	.057	.062	.065	.072	.078	.084	.090	.097	.100	.105	.105	.120	.130		

20-Inch Crane Round Boiler—Performance Data

Number of Boiler	Output in Equivalent Direct Radiation Sq. Ft.	280 448 672	300 480 720	320 512 768	340 544 816	360 576 864	380 608 912	400 640 900	420 672 1008	440 704 1056	460 736 1008	480 768 1152	500 800 1200	520 832 1248	540 864 1296	560 896 1344	580 928 1392	600 960 1440	640 1024 1536	680 1056 1584	680 1056 1632	Fuel Cap. Fuel Avail.	Stack Size and Height
S-204 W-204 Grate Area 2.18 Sq. Ft.	Firing Period—Hours	24.3	23.6	22.6	21.2	20.4	19.9	18.8	17.7	17.1	16.2	15.0	14.3	13.5	13.1	12.5	11.6	11.0	10.7	10.0	9.7	240# Cap. 180# Avail.	8x8 Ins. 30 ft. High
	Boiler Efficiency—%	72.5	72.2	72.2	72.0	72.0	71.7	71.5	71.1	70.9	70.5	70.1	70.1	69.9	69.9	69.8	69.2	69.0	68.8	68.4	68.8		
	Stack Gas Temp.—°F	425	450	480	500	530	550	575	600	625	650	675	700	730	750	775	805	830	880	900	900		
	Combustion Rate	3.40	3.65	3.90	4.15	4.40	4.70	4.95	5.21	5.46	5.75	6.03	6.30	6.60	6.92	7.12	7.42	7.73	8.28	8.55	9.7		
	Draft—Ins. of Water	.030	.034	.037	.042	.047	.053	.060	.064	.068	.072	.077	.082	.087	.092	.098	.105	.110	.125	.135			
S-205 W-205 Grate Area 2.18 Sq. Ft.	Firing Period—Hours	24.7	23.1	21.5	20.4	19.2	18.1	17.0	16.2	15.4	14.7	14.1	13.5	12.9	12.4	11.9	11.5	11.0	10.3	10.0	9.7	240# Cap. 180# Avail.	8x8 Ins. 35 ft. High
	Boiler Efficiency—%	74.1	74.0	73.9	73.6	73.4	73.0	72.8	72.5	72.3	72.0	72.0	71.8	71.6	71.4	71.2	71.1	70.8	70.4	70.3	70.3		
	Stack Gas Temp.—°F	340	355	370	385	400	415	430	445	460	480	495	510	530	550	560	575	590	625	640	655		
	Combustion Rate	3.35	3.57	3.83	4.05	4.30	4.57	4.84	5.10	5.34	5.60	5.85	6.13	6.38	6.66	6.92	7.20	7.46	8.00	8.25	8.52		
	Draft—Ins. of Water	.028	.031	.033	.035	.038	.041	.044	.047	.049	.053	.057	.060	.064	.068	.072	.082	.090	.100	.105	.110		
S-206 W-206 Grate Area 2.18 Sq. Ft.	Firing Period—Hours	23.6	22.3	21.1	20.0	19.0	18.0	17.5	16.6	15.8	15.1	14.5	13.9	13.3	13.1	12.1	11.8	11.4	10.7	10.3	10.0	240# Cap. 180# Avail.	8x8 Ins. 35 ft. High
	Boiler Efficiency—%	75.1	75.0	75.0	74.8	74.8	74.7	74.5	74.4	74.3	74.1	74.1	73.9	73.8	73.7	73.5	73.4	73.0	72.7	72.5	72.5		
	Stack Gas Temp.—°F	305	320	330	345	355	370	380	395	410	420	430	430	445	460	470	485	495	520	535	550		
	Combustion Rate	3.50	3.70	4.00	4.24	4.48	4.72	4.97	5.22	5.48	5.70	5.95	6.20	6.46	6.70	6.97	7.24	7.46	7.75	8.00	8.25		
	Draft—Ins. of Water	.027	.030	.030	.033	.039	.045	.045	.049	.050	.057	.060	.063	.069	.071	.075	.082	.087	.088	.092	.096		

Table 25. Crane Sectional Boiler for Oil Burning—Performance Data

Output in Equivalent Direct Radiation Sq. Ft.

Number of Boiler	Item	100 / 160 / 24	175 / 280 / 42	250 / 400 / 60	325 / 520 / 78	400 / 640 / 96	475 / 760 / 114	550 / 880 / 132	625 / 1000 / 150	700 / 1120 / 168	775 / 1240 / 186	850 / 1360 / 204	925 / 1480 / 222	1000 / 1600 / 240	1075 / 1720 / 258	1150 / 1840 / 276	Heating Surface	Stack Size and Height
1-5-SO	Heat Transmission Rate	1043	1826	2608	3391	4174	4957	5739									23.0 Sq. ft.	8x8 30 ft.
1-5-WO	Minimum Burner Capacity	1.6	2.9	4.2	5.6	6.9	8.4	9.9										
1-6-SO	Heat Transmission Rate		1522	2174	2826	3478	4130	4783	5435	6087							27.6 Sq. ft.	8x8 30 ft.
1-6-WO	Minimum Burner Capacity		2.8	4.1	5.5	6.8	8.2	9.7	10.4	12.6								
1-7-SO	Heat Transmission Rate				1863	2422	2981	3540	4099	4658	5217	5776					32.2 Sq. ft.	8x8 30 ft.
1-7-WO	Minimum Burner Capacity				4.1	5.4	6.7	8.1	9.6	10.3	12.5	14.0						
1-8-SO	Heat Transmission Rate					2120	2609	3098	3598	4076	4565	5054	5543	6033			36.8 Sq. ft.	8x8 30 ft.
1-8-WO	Minimum Burner Capacity					5.4	6.7	8.1	9.5	10.1	12.3	13.7	15.2	16.7				
1-9-SO	Heat Transmission Rate						2319	2754	3188	3623	4058	4493	4928	5362	5797		41.4 Sq. ft.	8x12 30 ft.
1-9-WO	Minimum Burner Capacity						6.6	8.0	9.4	10.0	12.1	13.5	15.0	16.5	18.0			
1-10-SO	Heat Transmission Rate						2478	2870	3261	3652	4043	4435	4826	5217	5609	6000	46.0 Sq. ft.	8x12 35 ft.
1-10-WO	Minimum Burner Capacity						7.9	9.3	9.9	12.0	13.4	14.8	16.3	17.7	19.3	20.8		

Table 26. Crane Sectional Boiler for Oil Burning—Performance Data

Output in Equivalent Direct Radiation Sq. Ft.

Number of Boiler	Item	400 / 640 / 96	525 / 840 / 126	660 / 1040 / 156	775 / 1240 / 188	900 / 1440 / 216	1025 / 1640 / 246	1150 / 1840 / 276	1275 / 2040 / 306	1400 / 2240 / 336	1525 / 2440 / 366	1650 / 2640 / 396	1775 / 2840 / 426	1900 / 3040 / 456	2025 / 3240 / 486	2160 / 3440 / 516	2275 / 3640 / 546	2400 / 3840 / 576	2525 / 4040 / 606	2660 / 4240 / 636	Heating Surface	Stack Size and Height
2-6-SO	Heat Transmission Rate	1920	2520	3120	3720	4320	4920	5520	6120												50.0 Sq. ft.	8x12 30 ft.
2-6-WO	Minimum Burner Capacity	6.6	8.8	11.0	13.3	15.7	18.1	20.4	23.0													
2-7-SO	Heat Transmission Rate		2161	2676	3190	3705	4220	4734	5249	5763											58.3 Sq. ft.	12x12 30 ft.
2-7-WO	Minimum Burner Capacity		8.7	10.9	13.1	15.5	17.7	20.0	22.7	25.2												
2-8-SO	Heat Transmission Rate			2342	2793	3243	3694	4144	4595	5045	5495	5946									66.6 Sq. ft.	12x12 35 ft.
2-8-WO	Minimum Burner Capacity			10.8	13.0	15.3	17.5	19.7	22.4	24.8	27.0	29.8										
2-9-SO	Heat Transmission Rate				2480	2880	3280	3680	4080	4480	4880	5280	5680	6080							75.0 Sq. ft.	12x12 35 ft.
2-9-WO	Minimum Burner Capacity				12.9	15.2	17.4	19.5	22.0	24.8	26.8	29.3	32.0	34.3								
2-10-SO	Heat Transmission Rate					2593	2953	3313	3673	4034	4394	4754	5114	5474	5834						83.3 Sq. ft.	12x12 40 ft.
2-10-WO	Minimum Burner Capacity					15.1	17.3	19.4	21.8	24.2	26.5	28.9	31.6	34.0	36.6							
2-11-SO	Heat Transmission Rate						2686	3013	3341	3668	3996	4323	4651	4978	5306	5633	5961				91.6 Sq. ft.	12x16 40 ft.
2-11-WO	Minimum Burner Capacity						17.2	19.2	21.6	24.0	26.3	28.7	31.1	33.6	36.1	38.9	41.0					
2-12-SO	Heat Transmission Rate							2760	3060	3360	3660	3960	4260	4560	4860	5160	5460	5760	6060		100 Sq. ft.	12x16 40 ft.
2-12-WO	Minimum Burner Capacity							19.0	21.5	23.8	26.1	28.3	30.9	33.3	35.8	38.3	40.5	43.3	45.5			
2-13-SO	Heat Transmission Rate								2825	3102	3380	3657	3934	4211	4488	4765	5042	5319	5596	5873	108.3 Sq. ft.	12x16 40 ft.
2-13-WO	Minimum Burner Capacity								21.4	23.8	26.0	28.3	30.7	33.1	35.6	37.8	40.2	42.7	45.0	47.7		

Select a boiler at the transmission rate desired for the total of the actual radiation load plus piping load plus "pick up" load. Minimum Burner Capacity=Minimum pounds of oil burned per hour.
Heat transmission rate=B.t.u. transferred per hour per sq. ft. of heating surface. Minimum Burner Capacity=Minimum pounds of oil burned per hour.

Example 1. To choose a steam boiler for hard coal, with total equivalent direct radiation load of 375 square feet, turn to Table 22. Locate figure 400 in italics, next higher than 375. Follow down the 400 column to data enclosed between heavy vertical lines. Indicated boilers are Nos. 1-6-S and 1-7-S and 1-8-S. The one chosen depends on firing period and combustion rate desired.

Example 2. To choose hot water boiler for hard coal, with total load of 1600 square feet radiation. Turn to Table 23. Locate figure 1640 in bold Roman type at top of column. Follow down column to data enclosed between heavy vertical lines. Indicated boilers are Nos. 2-8-W and 2-9-W and 2-10-W. The one selected depends on firing period, etc.

Tables 24 to 26 are used in like manner.

Soft Coal Burning. The performance data given for boilers is based on fuel having a calorific value of 12,500 B.t.u. per pound. We find that if the fuel value varies from this figure an allowance of 1% per 100 B.t.u. should be made. For example, if the fuel to be used has a value of 12,000 B.t.u., instead of 12,500 B.t.u., the output required of the boiler should be increased 5%. On the other hand, if the fuel should have a value of 14,000 B.t.u. per pound the required output should be decreased 15%.

To explain further, assume the following case:

Radiator load..............................	500 square feet
*Piping load................................	100 square feet
Fuel to be used............................	14,000 B.t.u. per pound
Total equivalent radiator load................	600 square feet
Less 15% fuel factor........................	90 square feet
Equivalent boiler output for 12,500 B.t.u. fuel..	510 square feet

This is the proper figure for output in equivalent direct radiation to be used for selecting a boiler from rating tables or performance data.

Example. To simplify and quicken the calculations given above, a condensed method is used, Fig. 23. Determine the B.t.u. value of the fuel to be used. Where the horizontal line denoting the fuel value intersects the diagonal it also intersects a vertical line, which represents the percentage by which the boiler size may be decreased.

14,500 B.t.u. per pound coal is to be used.

Steam radiation and piping is 550 square feet.

According to Fig. 23, on 14,500 B.t.u. coal, boiler size may be reduced 20%.

Therefore: 550 square feet of steam radiation, less 20% = 440 square feet, total load.

In Table 22 with this fuel, we find that a boiler No. 1-7-S could be selected, whereas with 12,500 B.t.u. fuel, a No. 1-8-S would be used ordinarily.

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*This amount can be estimated at from ⅛ to ¼% of radiator load.
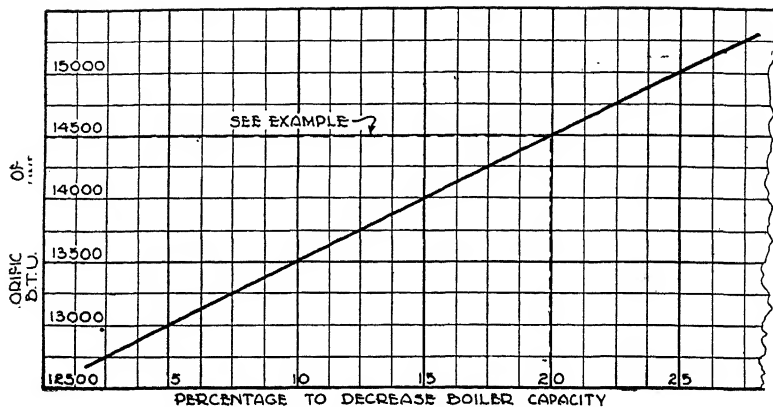
Fig. 23. Condensed Method Chart (See Example, page 87).
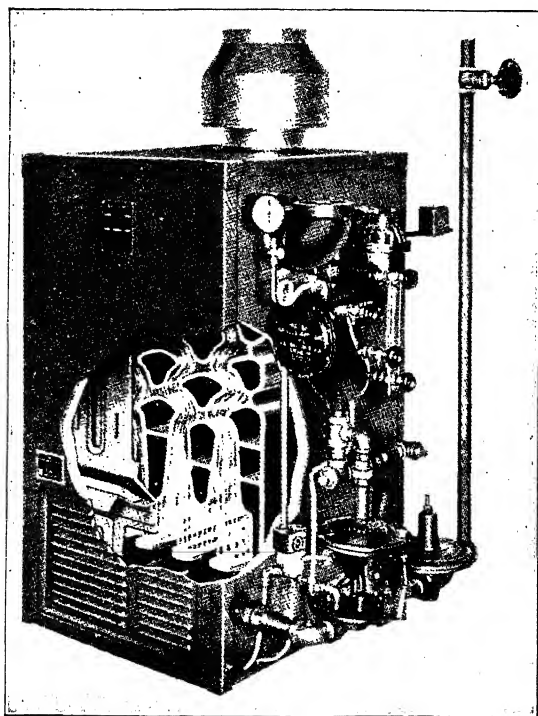
Fig. 24. Gas-Fired Boiler
Courtesy of Bryant Heater Co.

Summary. In this section two methods of determining boiler sizes have been given with illustrative examples. A short summary of each method follows:

The first method can be called the B.t.u. method because, to determine boiler sizes by that method, it is necessary to know the heat loss for the entire structure in B.t.u. per hour. The calculation of heat loss is explained in Chapter II. Knowing the total heat loss in B.t.u. per hour, the boiler size can be determined by the method illustrated under the head of "Sectional Boilers" in the first part of this section. Then in Table 22, for example, the correct size of boiler can be selected from the left-hand column of the table.

The second method can be called the equivalent direct radiation method because, to determine the correct size of a boiler, it is necessary to know the total equivalent radiator load. The calculation of the radiator load can be done by either method A or B explained in the chapter on "Radiators." In method A the total radiator requirements are calculated directly without calculating heat loss per hour. In method B it is necessary to calculate the heat loss in B.t.u. per hour and from that determine the radiator requirements. In both A and B the piping load must be added. Then, from Table 22, for example, the correct boiler size is selected as per illustrative example on page 87.

Gas=Fired Boiler. Fig. 24 is a cutaway view of a typical gas-fired boiler and illustrates the burners and heat travel. Such a boiler can be used for steam, hot water, or vapor installations. The water tube sections are of cast iron, and are designed to provide a staggered gas travel and ribbed to present a maximum area of heat absorbing surface.

For steam heating the boiler has the following trim:

Boiler control (steam pressure governor and low water cut-off); solenoid valve, throttling control valve, combination thermostatic pilot and escapement burner, pop safety valve (set 15 pound); gas pressure regulator, labeled shut-off valve, compound pressure and vacuum gauge and siphon, water level gauge, metal jacketed cover of blue crackle finish, draft hood, drain cocks.

For hot water the boiler has the following trim:

Gas-actuated limit control, combination thermostatic pilot and escapement burner, solenoid valve, diaphragm snap valve, gas pressure regulator, labeled shut-off valve, altitude gauge and thermometer, metal jacketed cover of blue crackle finish, draft hood, drain cocks.

Table 27. Ratings, Heating Surfaces, and Water Capacity for
Bryant Type Gas-Fired Boiler

Boiler No.	Available B.t.u. per Hour (Output)	A.G.A. Rating Sq. Ft.		Supplies Sq. Ft. of Direct C. I. Radiation *		Gals. per Hour 60° Rise	Boiler Horsepower	Sq. Ft. of Heating Surface	Water Capacity to Water Line, Gal.	Size Flow and Return Tappings, In.
		Steam	Water	Steam	Water					
3	158,400	660	1060	425	685	317	4.73	41.7	17.8	4
4	211,200	880	1410	565	920	422	6.30	55.1	22.2	4
5	264,000	1100	1760	710	1160	528	7.88	68.5	26.2	4
6	316,800	1320	2110	860	1410	634	9.55	81.9	31.0	4
7	369,600	1540	2460	1010	1665	739	11.00	95.3	35.4	4
8	422,400	1760	2820	1160	1935	845	12.60	108.7	39.8	4
9	475,200	1980	3170	1320	2190	950	14.20	122.1	44.2	4
10	528,000	2200	3520	1470	2445	1055	15.70	135.5	48.6	4
11	580,800	2420	3870	1640	2695	1160	17.30	148.9	53.0	4
12	633,600	2640	4220	1790	2960	1270	18.90	162.3	57.4	4
13	686,400	2860	4580	1965	3230	1370	20.50	175.7	61.8	4

*Selection factors providing for piping loss and starting load allowances are those recommended by the American Gas Association.

Oil-Fired Boiler. Fig. 25 shows a sectional view of an oil-fired boiler. It is a low-pressure unit designed to function with either radiator or air-conditioning systems. This differs in design from other boilers which incorporate burner units such as pot burners, gun-pressure burners, and rotary-type burners.

The pot-burner represents the first state of refinement beyond the ordinary wick type of lamp or range oil burner. Oil is burned directly from a liquid surface or by dripping on a hot metal surface. Forced draft is generally necessary because only a low pressure fan provides combustion air. The oil, because no atomization is provided, is No. 1 fuel oil or kerosene. It is ignited by a gas pilot unless the burner operates continuously.

In the gun-pressure burner, oil is put under a relatively high pressure (70 to 125 pounds per square inch) and forced through a small orifice to break it up into minute particles. Air is supplied by a pump or blower and regulation is attempted in order to provide a concentric stream of air to mingle with the oil spray. With the degree of atomization obtained in these burners, electric ignition is used commonly and either No. 2 or No. 3 oil is the fuel. The burner unit is installed in the ashpit, which is bricked to form a primary combustion chamber, and the flame is directed toward the back of the furnace.

Another example of the atomizing burner is the rotary type.

This consists of a small high-speed motor driving a spinner which breaks up the oil by whirling it out radially. The rotating unit usually includes a fan to provide combustion air, the fan being built as a part of the spinner or as a separate element mounted on the same shaft. Most of the rotary burners employ a ring of refractory
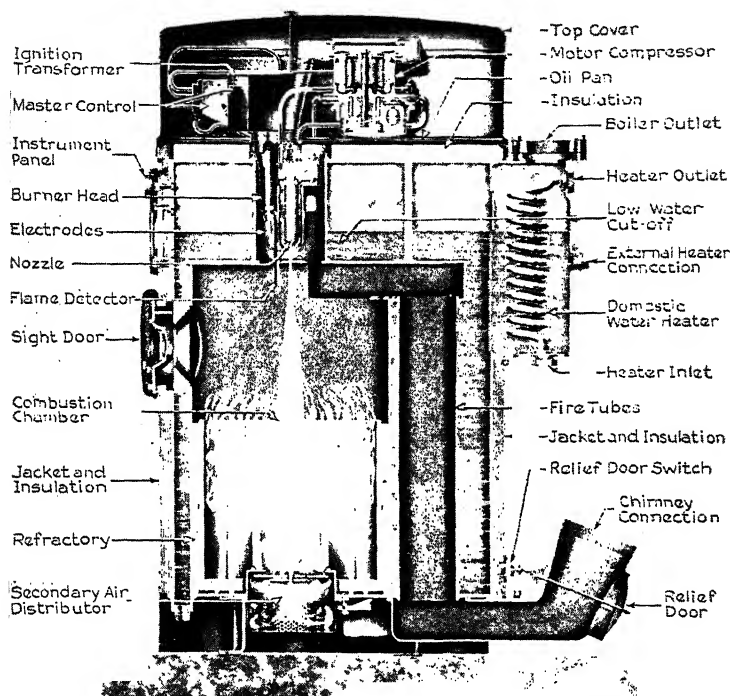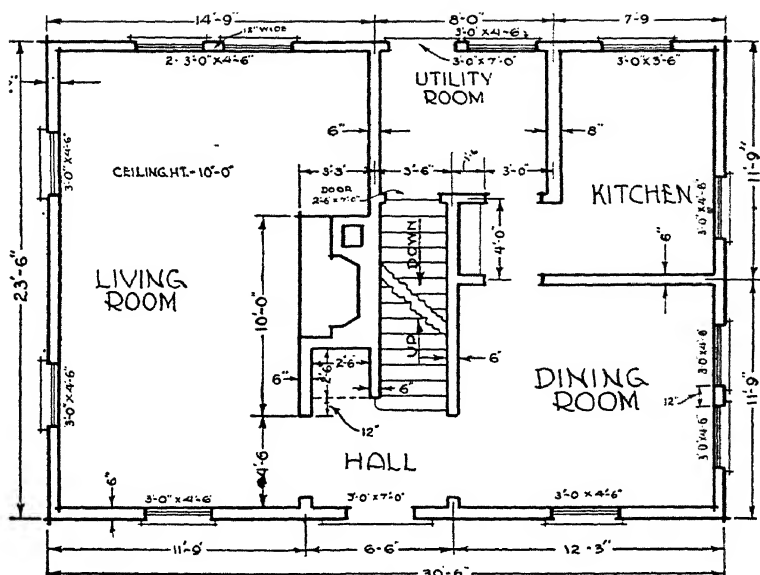

Fig. 25. Sectional View of Oil Furnace
Courtesy of General Electric Co.

material or highly tempered alloy steel around the inside of the combustion chamber. When the refractory ring becomes thoroughly heated, it vaporizes the oil particles thrown on it by the spinner. Most rotary burners use electric ignition, although in this type more dependable results can be obtained with gas ignition.

This burner provides air at three different points, and the breaking of the oil is carried on in two steps rather than in one. The initial action of air colliding with oil is performed at a pressure between 12 and 15 pounds in a small mixing chamber within the nozzle tip.

Under this low pressure the oil-air mixture is discharged through an orifice into the combustion chamber, where the pressure is immediately released. This release of pressure reduces to a minimum the pushing of hot gases through the boiler and up the stack with a consequent loss of usable heat. The process of impact expansion atomization takes place at the orifice where each drop of oil, as it passes through the orifice, is broken up into millions of tiny particles, giving a veritable fog of oil. This complete atomization makes it possible to burn a low-grade, low-cost, high-heat-content fuel oil.

Fig. 26. Small House Floor Plan

Primary air from the motor compressor is blown around the nozzle tip collar to cool the tip and to aid the electric spark ignition. Secondary air is introduced at the bottom through a refractory nozzle. The oil spray directed from above is centered and burned by this combustion air. Combustion is such that the flame turns up and floats above the refractory nozzle, insuring complete burning of the oil.

All heat radiated from the flame is diverted into the boiler walls and, as 96% of the boiler is water-backed, heat is picked up effectively by the water. Steam generated in the boiler may be used in a radiator system or carried to the heating coils of an air conditioner.

This boiler contains provisions for heating a domestic water supply. The water heater is shown in Fig. 25. The coils of the heater are surrounded by hot water which heats the water within the coils.

FIRST FLOOR PLAN

Fig. 27. First Floor Plan
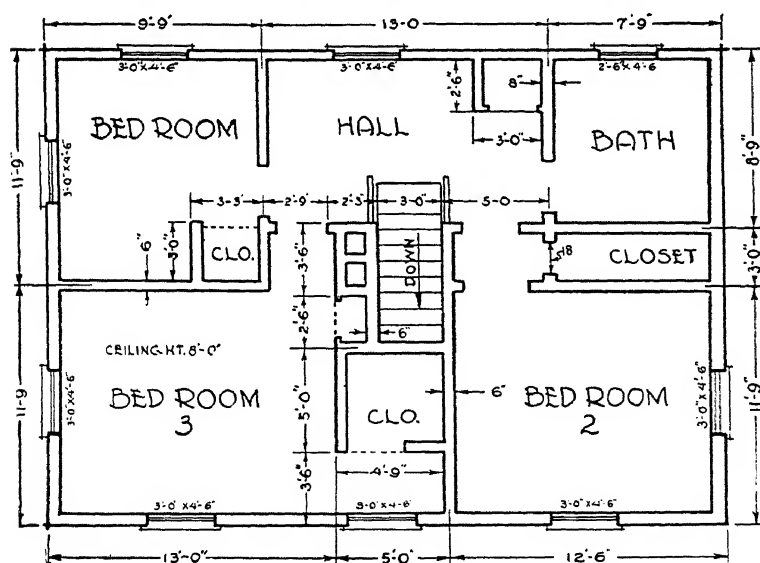
SECOND FLOOR PLAN

SCALE

REDUCED SCALE $\frac{1}{4}$" = 1' - 0"

Fig. 28. Second Floor Plan

Table 28 shows ratings for two typical oil-fired boilers and includes domestic hot water supply.

Table 28.   Ratings for Oil=Fired Boilers

	TYPE	
Total Output	LA-4	LA-5
Maximum, at boiler outlet, B.t.u. per hr	133,000	275,000
Equivalent steam radiation, sq. ft	555*	1145*
Equivalent hot water radiation, sq. ft	885*	1835*

* These figures are total boiler load and, in general, the permissible standing radiation will be less than this. Steam radiation based on 240 B.t.u. per hr. per sq. ft. Hot water radiation based on 150 B.t.u. per hr. per sq. ft.

Domestic Hot Water	LA-4	LA-5
Heating Capacity, B.t.u. per hr.		
Steam System	18,000	18,000
Hot Water System	25,000	25,000
Heating Capacity, gal. per day		
Steam System	175 to 400†	
Hot Water System	175 to 500†	

† Depending on size and location of storage tank.

Power Consumption	LA-4	LA-5
At maximum oil rate, watts	200	220
At minimum oil rate, watts	195	210

On hot water models add 90 watts for water circulator.

Note: Complete rating tables for all types of boilers shown can be secured from the manufacturers.

PRACTICE PROBLEMS

1. In the section on "Radiators," Chapter VII, there are two practice problems, Nos. 1 and 2 on page 163, for determining the number of square feet of steam radiation for the various rooms in Figs. 69 and 70. After reading Chapter VII and finding the answers to those problems, determine the numbers of the steam boilers that could be used to supply the necessary equivalent square feet of radiation. Use the equivalent direct radiation method of boiler selection. Assume soft coal having a calorific value of 14,500 B.t.u. Use round or sectional boiler as thought best.

2. Refer to Fig. 26. Assume temperature and structural specifications and calculate the heat losses. Then determine the size of grate necessary for a steam boiler to supply the necessary heat. Assume soft coal with a calorific value of 12,500 B.t.u. Which type of boiler would be best, round or sectional, when considering cost, appearance, etc.? Specify the boiler number.

3. Refer to Figs. 27, 28, and 29. Assume temperature and structural data. Using method *A* of the section on "Radiators" and the equivalent direct radiation method of this section, determine the number of the steam boiler, either sectional or round, best suited to supply the required equivalent direct radiation. Assume fuel having a calorific value of 15,000 B.t.u.

*Stoker=Fired Boilers. To illustrate the principles of this discussion, Ideal automatic coal burning boilers are used.

All boilers for automatic coal burning carry the identification letter C. In ordering boilers for automatic coal burning the C is added to the regular catalogue number. For example:

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* Data Courtesy of American Radiator Co., New York.

(a) A four-section No. 21 steam boiler will be designated as an SC-421; a water boiler, WC-421. (See typical manufacturers' ratings in Table 29, page 102.)

Rating of Stoker Boilers. Ratings on all boilers are shown as the actual installed radiation that the boilers will handle. No allowance need be made for normal mains, returns, and risers or pick-up load. In establishing the ratings an allowance of $33\frac{1}{3}\%$
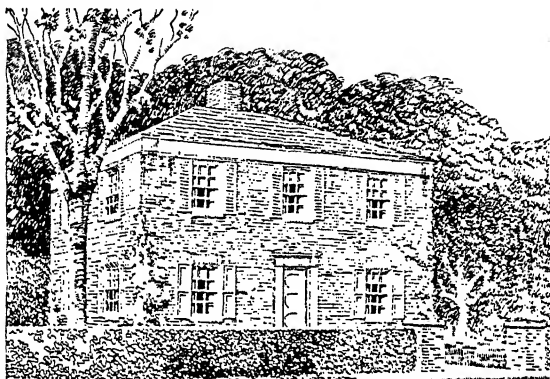
Fig. 29. Elevation

has been made for this purpose. This usually is a safe figure and one generally used throughout the stoker industry.

The amount of radiation in mains, returns, and risers, however, varies widely on different installations. For example, in large buildings where the length is much greater than the width, necessitating long mains and returns; in two- or three-family houses, where one boiler serves the entire heating plant; or in garages where there are long mains which may or may not be covered, the usual $33\frac{1}{3}\%$ allowance does not apply. Therefore, caution should be used in determining an adequate allowance.

While ratings shown are the actual installed radiation boilers will handle, any guarantee for satisfactory heating of buildings must be based on a sufficient amount of radiation being installed.

Coal burner minimum capacities are based on actual installed radiation plus an allowance of $33\frac{1}{3}\%$ for mains, returns, and risers. This applies when coal of 12,500 B.t.u. per pound is being burned at

*Stokers are explained starting on page 289.

the combined stoker and boiler-operating efficiency that will insure a flexible, easy operating unit under practical operating conditions with minimum fuel and power costs. The coal burner minimum capacity in pounds of coal per hour is the amount of coal per hour the burner selected must burn to develop the rating shown.

The efficiencies that have been used are not the maximum efficiencies stokers and boilers are capable of producing. In fact, higher efficiencies usually are obtained.

When other than 12,500 B.t.u. coal is to be burned, adjustments must be made in the minimum coal burner capacity. For example, in an SC-421 boiler, if coal of 10,000 B.t.u. per pound is to be burned, the minimum coal burner capacity will be 25 pounds of coal per hour instead of 20 pounds.

If 14,000 B.t.u. coal is to be burned, the minimum coal burner capacity will be 18 pounds per hour, approximately, instead of 20. The coal burner should burn sufficient coal to allow for a reasonable margin of safety, usually from 15 to 20% above minimum coal burner capacity required by the boiler.

When other than 12,500 B.t.u. coal is to be burned, adjustments must be made in the minimum coal burner capacity. For example, in an SC-421 boiler, if 10,000 B.t.u. coal is to be burned, the minimum coal burner capacity will be 25 pounds of coal per hour instead of 20 pounds. If 14,000 B.t.u. coal is to be burned, the minimum coal burner capacity will be 18 pounds of coal per hour, approximately, instead of 20 pounds.

Leading coal burner manufacturers do not recommend the selection of coal burners based on the absolute maximum pounds of coal per hour the burner will burn. The coal burner should burn sufficient coal to allow for a reasonable margin of safety, usually from 15 to 20% above the minimum coal burner capacity required by the boiler.

Selecting Stoker Boilers. Before giving an explanation of the selection of stoker boilers we shall discuss briefly some stoker characteristics.

Effective Grate Area. Effective grate areas are different under hand-firing conditions than under stoker-firing conditions. With a hand-fired boiler, the effective grate area usually is the entire area of the grate surface. With a stoker-fired boiler, the effective grate area is the area in the fire box over which the stoker is capable of distributing and burning coal.

With bituminous stokers installed in small round or small square and rectangular boilers, the stoker retort is usually placed in the center of the fire box

and the space between the outside of retort and boiler side walls is filled with refractory material. In this case, *the effective grate area is the entire area within the fire box.* With anthracite stokers installed in small round or small square and rectangular boilers the retort usually is placed in the center of the boiler, but no refractory material is used because space must be allowed between the outside of the retort and boiler side walls for ash to fall through. In this case *the effective grate area is the area of the retort itself.*

In long, rectangular boilers in which bituminous stokers are installed, we have an entirely different situation. For example, in an SC-36T11 water tube boiler under hand-fired conditions the effective grate area amounts to 15 square feet. With stoker firing this amount of grate surface would not be used, but a bridge wall would be constructed in the fire box approximately 3 feet from the inside of the front boiler wall. A retort approximately 14 inches wide and 20 inches long would be installed, and cast-iron dead plates used to fill in the distance between front of retort and front of boiler, sides of retort and sides of boiler, and rear of retort and bridge wall. In this case, the effective grate area would be 9 square feet, approximately, instead of 15 square feet as under hand-fired conditions.

Thus it is obvious that the amount of coal burned per hour per square foot of grate area under stoker-firing is greater than under hand-firing conditions. Under hand-firing conditions the coal burned per square foot of grate per hour usually would not exceed 6 or 8 pounds—except for morning pickup—while with stoker-firing it would amount to approximately 15.8 pounds per square foot.

Headroom and Combustion Space. Headroom and combustion space should not be confused. They are entirely different. Headroom is the distance between top of retort and boiler crownsheet, and is required not only for efficient combustion, but to prevent a torch-like flame from being directed at any spot in the crownsheet or sections of the boiler and producing a crack.

Combustion space is the number of cubic feet required in the combustion chamber for complete combustion of the fuel. This space must be sufficiently large to permit of a reasonably high B.t.u. release per cubic foot per hour. If an excessive heat release is attempted, we usually are confronted with a bottled heat condition in the boiler, which cannot do other than damage various parts. Front doors get excessively hot, refractory runs, boiler fronts warp, dead plates and tuyeres are damaged, and the job becomes unsatisfactory.

Minimum Headroom Heights Required for Stokers from Top of Retort to Crownsheet of Boiler

Stokers Burning—

0 to	20 pounds coal per hour	18 inches
20 to	30 pounds coal per hour	20 inches
30 to	40 pounds coal per hour	24 inches
40 to	100 pounds coal per hour	28 inches
100 to	200 pounds coal per hour	30 inches
200 to	300 pounds coal per hour	36 inches
350 to	500 pounds coal per hour	40 inches
500 to	750 pounds coal per hour	48 inches
750 to	1200 pounds coal per hour	60 inches

Average Diameter of Retort of Anthracite Ash=removing Stokers

0 to 20 pounds coal per hour..........15 inches plus 3 inches for Scraper Arm
20 to 35 pounds coal per hour..........18 inches plus 3 inches for Scraper Arm
35 to 50 pounds coal per hour..........22 inches plus 3 inches for Scraper Arm

Average Diameter of Retort of Anthracite Non=ash=removing Stokers

0 to 20 pounds coal per hour, 15 inches   20 to 35 pounds coal per hour, 18 inches
35 to 50 pounds coal per hour, 22 inches

Domestic Hot Water Supply Loads

When the domestic hot water heater and storage tank are connected to the boiler, add 1 square foot per gallon of storage tank capacity to the actual installed radiation boiler will handle on steam boilers, and 2 square feet per gallon of storage tank capacity on water boilers, before selecting boiler sizes.

When tankless heaters are installed, add 50 square feet steam radiation per bathroom or 80 square feet water radiation per bathroom to actual installed radiation before selecting boiler size.

TYPICAL EXAMPLES IN SIZING BOILER

The following examples show the simplicity of sizing boilers and automatic coal burners, using the figures shown in Table 29.

Example 1. Building has 600 square feet of actual installed steam radiation. A No. 21 SC-521 boiler would be used, and the coal burner must have a minimum coal burning capacity in pounds of coal per hour when burning 12,500 B.t.u. coal of 26 pounds.

Example 2. Building has 660 square feet of actual installed steam radiation and a 60-gallon hot water tank which is connected to the boiler. Adding 1 square foot per gallon to the actual installed radiation brings the total to 720 square feet, and in this case a No. 21 SC-621 boiler would be used, and the minimum coal burner capacity in pounds of coal per hour when burning coal of 12,500 B.t.u. would be 32 pounds.

Example 3. Building has 800 square feet of actual installed steam radiation. It contains two bathrooms and a tankless heater is to be used to heat service water. Adding 50 square feet of steam radiation per bathroom, or 100 square feet to the installed radiation of 800 square feet brings the total installed radiation to 900 square feet of standing radiation. Therefore, in this case an SC-721 boiler would be used, and the minimum coal burner capacity in pounds of coal per hour when burning coal of 12,500 B.t.u. would be approximately 38 pounds.

Boiler Protector. Fig. 30 shows a No. 855 Micro Water-Boy. This automatic control is a specific example of boiler protection.

Water enters feeder at A. The large scale space B is for the accumulation of pipe scale. C is a fine mesh monel metal filter screen. D is a scale agitator on the end of valve E. E is a nickel silver needle valve. F is a compound monel metal leverage. G is a copper float. H is an outlet tube. I is an equalizing line that connects to top open-

ing of water glass gauge. *J* is a union nut that holds assembly together. *K* is a removable bonnet on which valve seat and all leverage are mounted.

Water from feeder enters the low water cut-out at *1*. The float *2* moves upward, pushing cam lever *3* against monel metal lever *4*,
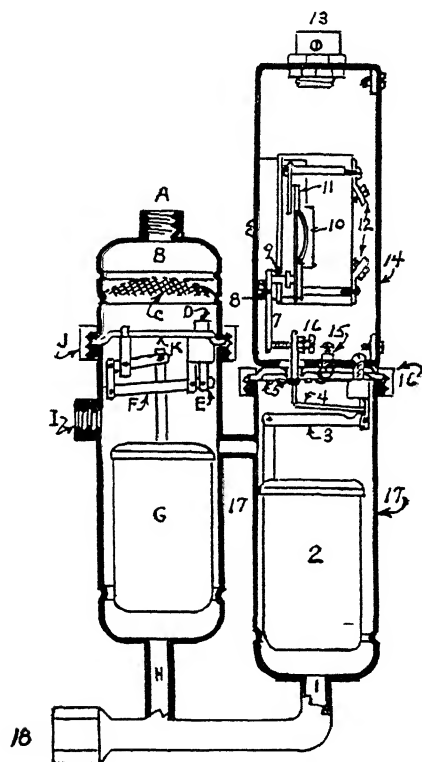
Fig. 30.   No. 855 "Micro" Water-Boy Boiler Protector
Courtesy of Maid-o'-Mist, Inc., Chicago

which is soldered to the diaphragm *5*. Lever *4* extends up into the switch box chamber and has an adjustment screw *6*, which contacts lever *7*, which is pivoted at *8*, and lever *7* contacts the switch push button *9*. This push button operates on spring *10*, which keeps the switch open until pressure pushes the spring past center and it snaps against contact. When pressure is released against push button *9*, the switch immediately snaps open. *11* indicates the silver contacts which are used to prevent pitting. *12* indicates the electric terminals.

Table 29. *Automatic Coal Burning Boilers

Size	Actual Installed Radiation Boiler Will Handle		Minimum Coal Burner Capacity Lbs. Coal per Hour	Boiler Heating Surface Sq. Ft.	Fire Box Dimensions		Distance from Floor to Crown-sheet of Boiler	Height of Base	Chimney	
	Steam	Water			Width	Length			Size Inches	Hgt. Feet
†C-421	510	815	20	42.3	23	22¼	*32¾	7⅛	8x 8	30
†C-521	710	1,136	27	54.7	23	30¼	*32¾	7⅛	8x 8	30
†C-621	910	1,456	34	67.1	23	38¼	*32¾	7⅛	8x12	35
†C-721	1,110	1,776	42	79.5	23	46¼	*32¾	7⅛	8x12	40
	Water									
WC-1705	350		10	18.2	17		39¼	13	8x 8	30
WC-1905	415		12	21.8	19		37½	13	8x 8	30
WC-2005	455		14	23.6	20		37	12½	8x 8	30
WC-2205	520		16	26.8	22		39½	13	8x 8	30
WC-2505	635		18	33.1	25		40¼	13¼	8x 8	35
WC-2805	775		22	40.3	28		40¾	13¼	10x10	35
WC-1706	410		12	21.4	17		39¼	13	8x 8	30
†C-36T 8	2,640	4,225	104	180.4	36	41½	38½	12¼	16x16	40
†C-36T 9	3,000	4,800	119	203.4	36	47½	38½	12¼	16x16	45
†C-36T10	3,340	5,345	132	226.4	36	53½	38½	12¼	16x16	50
†C-36T11	3,715	5,945	147	249.4	36	59½	38½	12¼	16x20	50
†C-36T12	4,055	6,485	160	272.4	36	65½	38½	12¼	16x20	55
†C-36T13	4,395	7,035	174	295.4	36	71½	38½	12¼	20x20	60
	Steam									
SC-1705	235		10	19.9	17		39¼	13	8x 8	30
SC-1905	285		12	23.7	19		37½	13	8x 8	30
SC-2005	315		14	26.0	20		37	12½	8x 8	30
SC-2205	345		16	28.8	22		39½	13	8x 8	30
SC-2505	430		18	35.8	25		40¼	13¼	8x 8	35

* This table is made up of portions taken from complete tables. Complete tables may be secured from the manufacturers.
† The C is preceded by S or W meaning steam or water as required.

13 is B-X connector. *14* is the switch box cover, and *15* is the pressure control. By screwing this screw down, a greater tension is put on diaphragm at *5*, to move it upward and force adjustment screw *6* away from its contact on lever *7*. Note that the diaphragm is knurled to carry a greater flexibility. The pressure regulator is set to cut out at 6 pounds, and cut in again at 1 pound. *16* is the assembly nut. *17* is the brass shell (the unit is built entirely from non-ferrous metal). *18* connects to the lower opening of the water glass gauge.

The automatic feeder shown in Fig. 30 is a safeguard against low water in boilers. Where automatic heat is applied to boilers, serious accidents have occurred when owners failed to watch the water level. In addition, such a control assures the system against excessive steam pressures.

CHAPTER VI

VENTILATING SYSTEMS

Ventilation can be accomplished by natural or by mechanical means. The natural principles depend on wind and temperature, while the mechanical principles are independent of everything but power supply. In this section both types will be discussed, with more detailed explanations given for mechanical principles because of the important part they play in air conditioning.

Natural Ventilation. Ordinary structures can be ventilated fairly well by horizontal ventilation. This is done by opening windows or doors on two sides of a room. Such ventilation depends on the presence of a wind to force the flow of air into one opening and out the other. Another way of ventilating is by opening windows from the top and bottom. Here air movement is caused by the warmer and lighter air rising to the top of the room, for example, where it is pushed out through the window by the cooler air coming into the room through the lower part of the window. This type of ventilation can be done better where inside and outside temperature difference exists, the outside temperature being the lower.

Natural ventilation may be supplied on a larger scale in structures where there are roof ventilators, louvers, etc., for the escape of air and where there are openings near the lower section of the building for incoming air. The functioning of such a system also depends on temperature differences. Thus, if temperature conditions are right, air will flow into the structure near the lower section and out through the roof ventilators. This system is not satisfactory because of the dependence on other natural conditions, which are not always right for proper functioning of the system.

Roof Ventilators. There are various types of these ventilators. Some are stationary and others change their position with the wind. The stationary type is constructed with a covering for protection against rain, snow, etc., and provides an easy escape of air despite wind directions. Other types have a vane which keeps the wind at the back of the openings and provides an increased flow of

air due to an aspirating effect. Fig. 31 shows a typical roof ventilator commonly used on barns or warehouses.

Mechanical Ventilation. Where ventilation or movement of air is brought about by mechanical means, such as fans and blowers, the results are positive and can be depended upon regardless of outdoor wind and temperature conditions. In addition, the desired
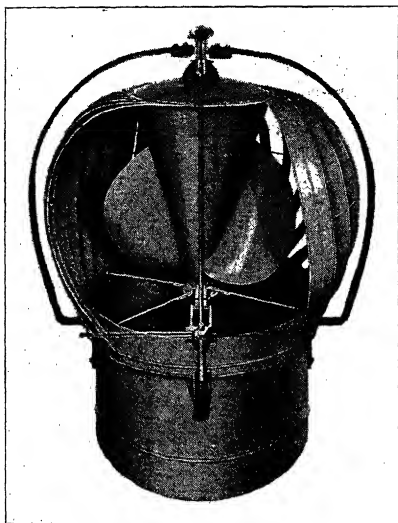
Fig. 31. Roof Ventilator
Courtesy of The Allen Corporation, Detroit
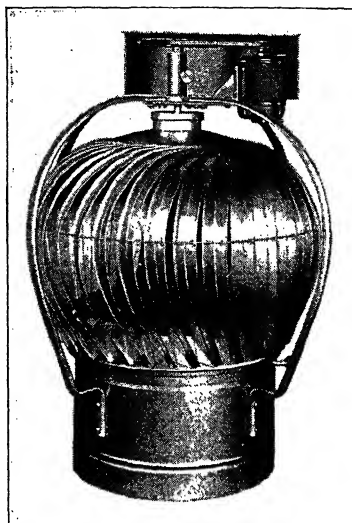
Fig. 32. Electro-Wind Turbine Ventilator,
Showing Motor
Courtesy of The Allen Corporation, Detroit

effects can be had without special architectural planning of windows, doors, and other openings.

Fig. 32 shows a type of electric roof ventilator where the powers of wind and electricity have been combined in an efficient ventilating apparatus. As a wind-operated ventilator, it is an efficient type. Electrically operated, it is a blower exhausting a definite amount of air regardless of temperature difference, wind velocity and direction, or other variable conditions. The worm geared-head motor and clutch are mounted on top of the rotor, leaving the throat of the ventilator clear of obstructions, such as fan blades, motors, drive shafts, etc.

This dual purpose exhauster or ventilator is designed to handle air-removal problems which require a large exhaust only a portion

of the time or under extraordinary conditions. When such situations arise, a switch or an automatic thermostatic control increases the exhaust. Under normal conditions, the wind-driven turbine functions efficiently and conserves power.

This ventilator requires no penthouse, as the window, regardless of wind direction or velocity, actually aids exhaust of the unit. It is easily installed on any standard ventilator base, and in most cases may be plugged into a lighting circuit for power connection.

To determine the size and number of ventilators required to handle a specific job, one must keep in mind a few simple rules. First of all, there are four primary factors affecting the capacities of roof ventilators which may be summed up as follows:

(a) The difference between the indoor and outdoor temperature, which is written $t - t_o$, is perhaps the greatest factor affecting the air flow. It is common knowledge that hot air is lighter than cold air and will rise to seek its level. The greater the temperature difference, the faster the rise. Consequently, an initial velocity is given the air, which aids the ventilator in its performance.

(b) The velocity of the wind (air in motion against any surface) will produce an area of lower pressure on the lee side of the surface and cause a partial vacuum in the ventilator head, accelerating the discharge of the air in the ventilator stack—a fact utilized advantageously in the design of all efficient ventilators.

(c) Height of ventilator above air inlets, or increased elevation, results in greater draft, so it follows that the higher the ventilator head above the air inlets to the room, the greater the exhaust.

(d) Area of ventilator stacks, that is, exhaust capacities, theoretically are directly proportional to the cross-sectional areas of their stacks, assuming, of course, that the ventilator heads are equally efficient. Inasmuch as circular areas vary directly as the square of their diameters, it follows that doubling the size of a ventilator will increase its area and exhaust capacity four times. For example, a 20-inch ventilator will exhaust roughly four times as much as a 10-inch ventilator of similar design.

The following points must be kept in mind when figuring ventilator requirements. Always be certain that provision is made for supply air to be drawn into rooms to take the place of air which is exhausted by the ventilators. Intake may be in the form of windows, doors, or other openings. If the room is closed most of the time, it will be necessary to install grilles or louvers, the free area of which should be twice the combined areas of the ventilator throats.

Properly designed ventilators do not down draft unless a tremendous amount of air is being removed from the same room by

mechanical means. Be sure there are no large electric fans or blowers exhausting great amounts of air from a point near the ventilator. One cannot expect natural ventilation to compete with powerful mechanical exhaust apparatus unless the latter is satisfied by ample provision for air supply. Combustion of coal or oil requires great quantities of air. A large boiler will create a starved air condition in a room which is not supplied with proper air intakes.

In instances where ventilators are required to remove large quantities of hot gases or steam, it is best to place a hood over the immediate source. This hood should be as low as possible and should project at least 2 feet around the vat or pot. If a hoist or conveyor makes it necessary for the hood to be placed a considerable distance above the source, this projection should be greater, depending upon the height.

. The hot gases rising into the hood must be drawn away quickly; therefore a large ventilator area is necessary to cope with the requirements. In such cases, the following empirical formula is useful: Allow 5,000 cubic feet per hour of exhaust capacity per square foot of hood area. For example, a hood 6x10 feet contains 60 square feet, which, multiplied by 5,000, gives 300,000 cubic feet per hour exhaust capacity. A glance at Table 30-A shows that the hood requires a 42-inch Multi-vane, or, as shown in Table 30-B, a 24-inch Electro-Wind Unit. This is the ideal adaptation for the Electro-Wind Unit, as the power is used only during such times as the gases are being generated at the source. The quenching of hot metals, for example, sends up clouds of steam at intervals, the electric drive being used only during those intervals.

Example: Assume a building $30 \times 60 \times 14$ feet high used as a garage. The building has no other structures near it and ample air comes in through and around the windows.

Solution: The cubical content of the building is $30 \times 60 \times 14 = 25,200$ cubic feet. For a garage where fumes from motors are present it is best to assume at least 12 changes of air per hour. Then, total air to be exhausted per hour is $12 \times 25,200 = 302,400$. The average effective area of a ventilator ranges from 10 to 15 feet radius or a unit approximately every 20 feet. The length of this building is 60 feet. Then $60 \div 20$ equals 3. Thus 3 ventilators would be required. Divide total air to be removed per hour by number of units required. $302,400 \div 3 = 100,800$ cubic feet per hour per ventilator.

Referring to Table 30-A it is found that a 20-inch ventilator exhausts 105,100 c.f.h. at 4 miles per hour wind velocity. Always use the 4-mile velocity table and select the ventilator size which just exceeds the exhaust required.

Table 30=A. Multi-Vane Turbine Ventilators

(Displacement Capacities)

Based on Height above Air Inlet—10 ft. $t-t_o=20°$ F.

Throat Diameter Inches	Free Air Displacement Cubic Feet per Hour		Throat Diameter Inches	Free Air Displacement Cubic Feet per Hour	
	Wind Velocity 4 Miles per Hour	Wind Velocity 8 Miles per Hour		Wind Velocity 4 Miles per Hour	Wind Velocity 8 Miles per Hour
6	10,100	12,200	20	105,100	125,600
8	17,300	21,600	24	149,000	185,000
10	26,500	32,500	30	225,000	272,000
12	38,600	46,200	36	281,000	330,000
15	54,000	69,000	42	324,000	414,000
18	85,200	102,100	48	360,000	473,000

Table 30=B. *Electro=Wind Turbine Ventilators

(Displacement Capacities)

Throat Dia. Inches	MOTOR		Rotor Speed R.p.m.	Air Displacement Cubic Feet Per Hour	
	R.p.m.	H.p.		Wind Driven Vel. 4 m.p.h.	Electric Driven
12	1725	1/10	216	38600	90000
15	1725	1/10	216	54000	126500
18	1725	1/8	216	85200	185100
20	1725	1/8	216	105000	225000
24	1725	1/6	173	149000	324000
30	1725	1/4	173	225000	487500

*Complete tables can be obtained from the manufacturer.

Kitchen and Toilet Room Ventilation. Kitchens require ample amounts of ventilation to clear away grease-laden smoke, odors, gases, etc. The principle of ventilation for kitchens and bathrooms is not to recirculate the air but to exhaust it completely. For this purpose ordinary exhaust fans are employed, as shown in Fig. 33. They are placed near the ceiling in walls or windows. Capacities of such fans, as needed specifically, can be had from manufacturers' catalogues.

The proper number of air changes per hour runs between 25 and 30. If the air in rooms is highly humid, cold air coming through open windows will cause condensation when it strikes the warm and humid inside air. In such cases warmed air must be supplied to the rooms in amounts similar to that exhausted every hour. In most cases the ordinary air coming in by infiltration is sufficient to supply the needs. Fig. 33 shows a typical fan used for exhaust work. Such
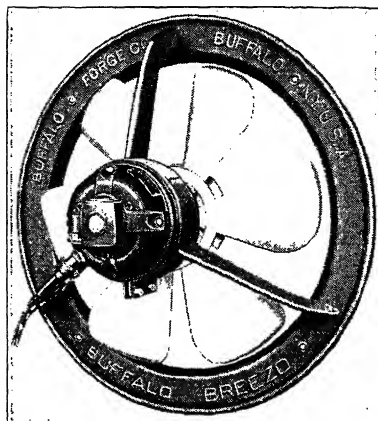
Fig. 33. Buffalo Breezo Fan
Courtesy of Buffalo Forge Co.

a fan can be mounted in ornamental casings, Fig. 34, or in plain metal casings for industrial use.

Office Ventilation. Offices or other enclosures can be well ventilated by the application of exhaust fans, as shown in Figs. 33 and 34. In this case good air is supplied by the fans, installed in a wall—preferably a rear wall to avoid marring the front elevation of the building. The fans carry off the stale air and circulate fresh air. The selection of fans is based on the area (cubic feet) of an enclosure in connection with the number of changes of air per hour.

Fig. 34. Application of Exhaust Fans to a Typical Enclosure—Fans Indicated by White Arrows
Courtesy of Buffalo Forge Company

Fans can be mounted in side walls or upper window spaces, set in wood or steel frames, or in transoms over doors or windows. They also can be mounted with vertical shafts to discharge air through
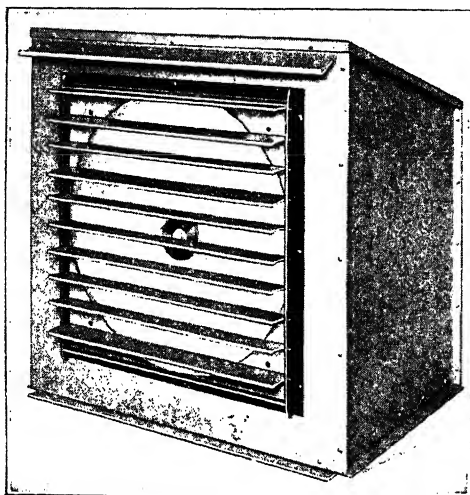
Fig. 35. Penthouse for Exhaust Fan
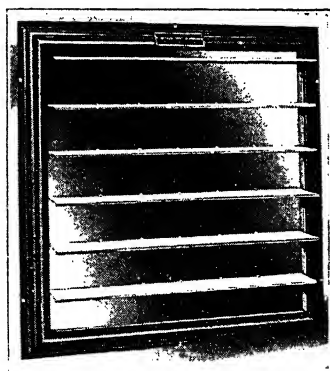
Fig. 36. Automatic Louvers
Courtesy of Buffalo Forge Company

flues. A desirable arrangement consists of a galvanized steel penthouse with motor-driven fan, automatic back-draft louver shutter, and hinged rainproof cover, shown in Figs. 35 and 36. These fans should be used with caution where duct runs are located, unless the duct runs are short and at least as large throughout as the fan

itself, because the resistance caused by ducts tends to cut down the air quantity. Advantage should be taken of the assistance which the prevailing wind will give. It is bad engineering practice, however, to use an exhaust fan for discharging air through a wall against which prevailing winds strike. If fans exhaust into a court having four walls, the wind direction need not be considered.
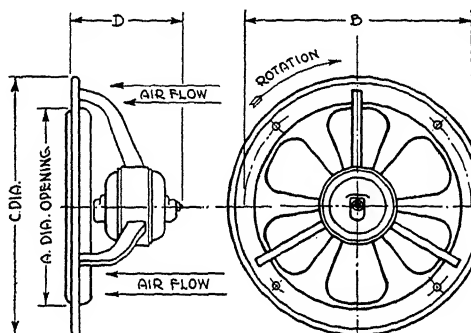
Fig. 37. Typical Over-all Dimensions for Exhaust Fans

Size		B	
8		10½	3½
12	12½	15½	6⅝
16	16½	22	9½
18	18½	24	9½
24	25	32	14½
30 AC-DC	31	40	18⅜
36 AC-DC	37	47	18⅛

Table 31 gives capacities of standard alternating- and direct-current fan units for typical fans. Fig. 37 gives typical over-all dimensions for the size of fans shown in Table 31.

Table 31. Capacities of Standard A.C. and D.C. Units

Size Inches	Speed R.p.m.	Capacity Cu. Ft.	Decible Rating	Size Inches	Speed R.p.m.	Capacity Cu. Ft.	Decible Rating
8	1500	500	30.0	18	850	1800	42.6
				18	1150	2400	49.0
12	750	460	28.0				
12	1150	700	35.7	24	670	3200	44.6
12	1725	1060	47.6	24	850	4000	48.5
16	850	1100	37.6	30	670	6200	56.6
16	1150	1500	44.6				
..	....	....	....	36	575	10000	56.0

Miscellaneous Ventilators. There are many types of ventilators of the unit or individual room type. Some have been made with the idea of cleaning the air as well as circulating fresh air.

A window type ventilator designed for use in connection with a single room is especially helpful for persons suffering from hay fever or similar ailments. This type of apparatus filters as well as circulates the air.

Many unit type air conditioners provide ventilation, besides cleaning, heating, and cooling the air. The principle of their ventilating process is a fan which produces a positive circulation of air throughout rooms or other enclosures. These units are explained in another section of the text.

Central Ventilating System. This system, as the name implies, is located in a central area and ducts are employed to distribute the air. The principle of central ventilating is made use of in the following three ways: the *split system*, the *heating and ventilating system*, and the *dividing system*.

Split System. In most buildings, such as office buildings, the heating is done by direct radiators, while the air supply or ventilation is done by a fan system which supplies air at approximately room temperature. Some systems have filters or washers or both and others merely supply untreated warmed or tempered air. How much conditioning is applied to the ventilating air depends upon conditions individual to each job and upon economic considerations. During summer months the systems can be utilized for supplying cooled air if a washer or other cooling apparatus forms part of the system.

Note: Air washers used as coolers are discussed in another section of this book.

While the split system has been used mostly in large types of buildings, it can be used as well in small buildings, single enclosures, and residences, as will be shown later.

One of the advantages of the split system is the ability to supply ventilation only when actually needed and thus create a savings in operating costs. In office buildings, the ventilation system need be operated only during working hours while the radiators keep up the heat to desired temperatures. In school buildings, the required temperature can be maintained by heat from the radiators, and the ventilating system can be started just before class time. Sections of buildings that are vacated for extended periods, such as storerooms, need not be ventilated but do require warming. The split system

makes that possible. In churches, where heat is maintained all week to protect the building against cold weather, the heat can be supplied by radiators and the ventilating system started just prior to the periods when the church will be occupied. In residences, one or more rooms can be supplied with heat only. Garages may be heated without being ventilated.

Following are discussions and explanations of control of a few typical fan systems operated by the split method.

Fig. 38 shows a tempered air installation of Aerofin* and fan
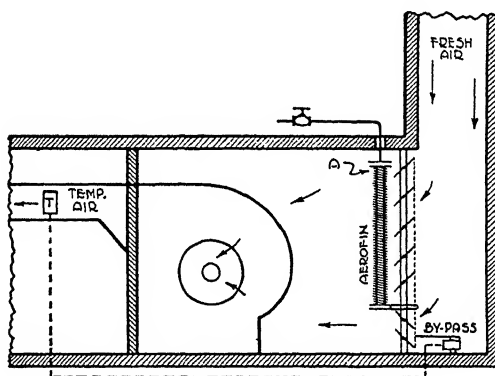
Fig. 38. Tempered Air for Ventilation System in Any
Climate (No Washer)

without an air washer. The structural work shown in the sketch is typical and is given to help the reader visualize the installation of the system. Fresh air is taken from the outside by way of a duct or opening. Generally the opening is faced with louvers as shown in Fig. 39. The air passes through the heating coil at A or through the by-pass, and is sucked into the fan† and forced throughout the building.

The thermostat is located in the discharge duct from the fan and controls louver mixing dampers covering the entire face of the tempering coil. The diaphragm motor operates these louver mixing dampers and also the by-pass damper beneath the tempering coils. These two dampers are hooked up together so the thermostatic control will provide the proper proportion of air through the tem-

*See page 385 for Aerofin explanation.
†See Chapter XIV, pages 249-272.

pering coil and through the by-pass. This produces a temperature of 64°F. to 70°F. in the tempered air for ventilation. The tempering group steam valve is not controlled. For the dampers, a graduated-action type thermostat* is preferable. A tempering group, as shown in Fig. 38, is only one unit deep in the direction of air flow. This battery generally contains tubes two rows deep, or the equivalent, for climates where the outside air temperature is figured at zero, and tubes three rows deep, or the equivalent, for climates where
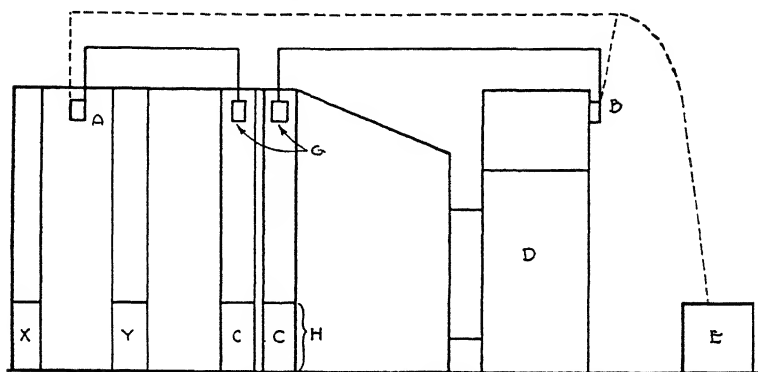
Fig. 39. System for Furnishing Filtered and Tempered Air for Ventilation; *A* and *B*, Thermostats; *C*, Heating Coil; *D*, Fan; *E*, Compressed Air; *G*, Sylphon Valve; *H*, By-Pass; *X*, Louver; *Y*, Filter

the outside air is figured at colder than zero. From a standpoint of economy in first cost, the arrangement in Fig. 38 is highly desirable for installation where no air washer is required. Such a system supplies ample air—although no humidification can be had nor summer cooling beyond *natural* cooling, as explained in the chapter on "Cooling Methods."

Fig. 40 shows an arrangement of coils for tempering any single enclosure, room or auditorium, where a single set of steam heaters is used. The room thermostat controls the duct thermostat. At *A* is a single row battery receiving outside air. It is controlled by a positive-acting thermostat located in the cold air inlet and set to turn steam on the battery at 35°F. or 40°F. It also regulates the steam on this outside single-row battery at all times when the outside temperature is colder than 35°F. or 40°F. This arrangement contemplates no air washer.

*See page 114, also Chapter XII.

In Fig. 39 is shown a rough sketch typical of the type of sketches that may be drawn before plans, as shown in Figs. 38 and 40, are made. Rough sketches are merely to aid in designing and visualizing a system.

The heating coils in Fig. 39 are similar to those already explained. They function so that the first coils heat incoming zero air to 35°F. and the second coils heat it from 35°F. to 70°F.
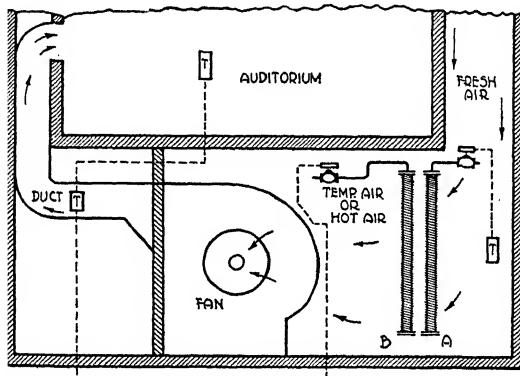
Fig. 40. Heating or Ventilating System for a Single Room in Climate of Zero or Above (No Washer)

A thermostat A of the positive-acting type is built in close to the louvers* where the cold outside air enters. The function of this thermostat is to control the "sylphon" valve on the first section of coils, and to close the valve should the incoming air reach a temperature of 35°F. At B the thermostat is of the graduated or modulated type and is arranged so that it will shut off the valve for the second section of heating coils if the temperature in the duct becomes higher than 70°F. This thermostat keeps air in the ducts at an even temperature of 70°F.

Zero air going through both sections of coils, as stated above, is heated to 70°F. The thermostat at A does not function until incoming air is 35°F. Thus, if outside air temperature rises above 0°F., the air in the duct at the location of B will rise, because the two sections of coils impart 70°F. to the incoming air. This means that if the incoming air is 10°F. the air at thermostat B would tend to be 70°F. However, thermostat B, being of the graduated-action

*See chapter on "Air-Conditioning Appliances."

type, operates to control the amount of steam admitted to the second section of coils and thus keeps the temperature 70°F.

A by-pass could be installed at H in Fig. 39 under the coils so that a part of the incoming air could be admitted to the fan without going through the coils. Thermostat B would then operate a damper motor which would control the opening and closing of the by-pass. Thermostat A operates as a positive-acting type and shuts off the first section of coils when the incoming temperature reaches 35°F. Thus, the second section of coils heats the 35°F. air to 70°F. The thermostat A will not change if incoming air reaches as high as 30°F. The second section of coils has no automatic control, therefore the air at thermostat B would be much too high in temperature except for the fact that this thermostat gradually opens the by-pass and the incoming air circulates around the coils and maintains a temperature of 70°F. A tempering ventilating system can be controlled in many ways, but the explanations already given are typical.

When tempered air ventilation requires an air washer and the system is the type that can be used in any climate, the reheater coils D and E in Fig. 41; E and F in Fig. 42; and C and D in Fig. 43 are omitted. Tempered air thermostats therefore would be replaced by a duct thermostat located in the fan outlet and there would be no by-pass or mixing dampers. The systems, shown in Figs. 41, 42, and 43 with the reheater coils omitted, would require an air washer and would apply to tempered air ventilation as follows: Fig. 41 to zero climate and warmer; Fig. 42 to climates colder than zero; and Fig. 43 to tempered air ventilation. The water heater in Fig. 43 would be located in the air washer tank. This arrangement would be entirely appropriate for use in any climate. Group C, Fig. 41, Group D, Fig. 42, and group B, Fig. 43 would in this case be one or two rows deep.

Heating and Ventilating System. This type of system supplies heat in amounts necessary to maintain desired temperatures in addition to the required ventilation. Thus, no direct radiators are necessary for heating, except under windows to prevent down drafts.

The air besides being heated may be filtered, washed, etc., according to requirements. The use of air washers is generally recommended to supply humidity during the heating season and some degree of cooling during the summer.

Fig. 41 shows an arrangement where there is a single-row battery of Aerofin Coils.* Group A is controlled by a positive-acting thermostat, located in the cold air inlet, which keeps steam turned on this single-row battery at all times when the outside temperature
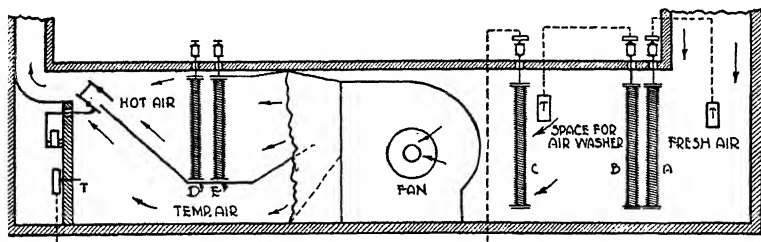
Fig. 41. Hot Blast Heating with Air Washer for Climates of Zero and Above

is $+35°$F. or colder. Group B is a single-row Aerofin battery controlled by a thermostat located in the space back of the air washer. Group C is a one-row retempering battery, controlled by a thermostat located in a tempering air chamber beyond the fan. Groups
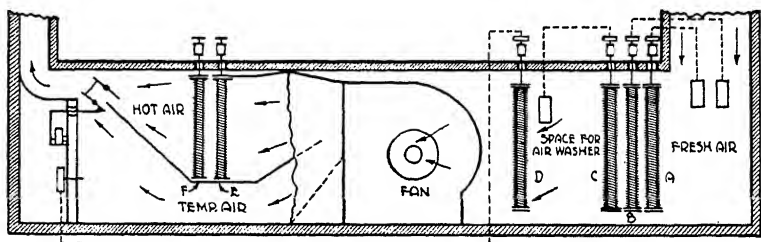
Fig. 42. Hot Blast Heating with Air Washer for Climates of Zero and Below

D and E may each be one or two rows deep as required. Fig. 41 is such a system as would be used for climates of zero or above.

Fig. 42 shows an arrangement for colder than zero climates, where group A is a single-row battery receiving the outside air. It is controlled by a positive-acting thermostat, located in the cold air inlet, which regulates the steam at all times when the outside temperature is $+35°$F. or colder. Group B is a second single-row battery controlled by another positive-acting thermostat, located in the cold air inlet, which regulates the steam at all times when the outside temperature is $10°$F. or colder. Group C is a third single-row bat-

*Coils explained in the chapter on "Air-Conditioning Appliances."

tery controlled by a thermostat which is located in the space back of the air washer. Group D is a single-row retempering battery located back of the air washer and in front of the fan. It is controlled by a thermostat located in the tempered air chamber beyond the fan. *Groups E and F may each be one or two rows deep as required to heat the given volume of air.

Fig. 43 shows a ventilating system supplied with a water heater. This system can be used in any climate and is similar to the systems shown in Figs. 41 and 42 except for the water heater located in the air washer tank. The water heater is controlled by a thermostat set at $+35°F$. It is located beyond the air washer in the direction of flow. This arrangement will provide a constant temperature of 35°F. with saturated air. The water heater takes the place of group B in Fig. 41 and of groups B and C in Fig. 42. Otherwise the arrangement of coils, with the number of rows deep and the manner of control, is the same as previously explained.

Dividing System. In this type of heating and ventilating, the fan system supplies in addition to the required ventilation about one-third of the actual heat needed to offset heat losses. The other two-thirds of the required heat is supplied by radiators. The process of designing such a system is similar to the previously explained systems, except that the coils must be capable of supplying one-third of the total heat needed. The controls are designed, as explained previously, so the air washers, filters, etc., can be used in the winter and the air washers or cooling coils in the summer.

Note 1: Filters are added to such typical fan systems as shown in Figs. 40 to 43, inclusive, and also in Figs. 39 and 51, where the design and operation of fan systems is further explained.

Note 2: See application example in chapter on "Air Washers."

Fig. 44 shows a cross section of an office building with the air conditioner and ducts in place. Any one of the three systems previously explained could be used here. If a split or dividing system is used, radiators should be installed. For an entire fan system, no radiators are necessary. Air can be discharged through the exit ducts to the outside or can be recirculated if an air washer or filter is used.

Fig. 45 shows an abbreviated drawing of the cross section of a

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*Selection of coil sizes explained, page 387.
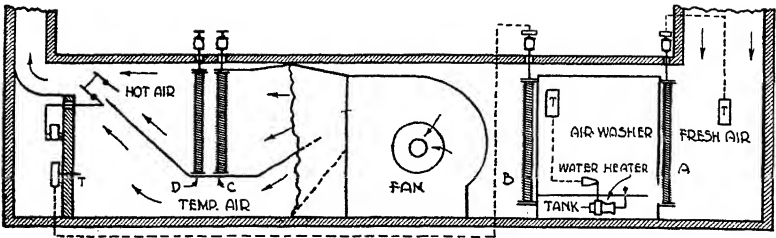
Fig. 43. Hot Blast Heating with Air Washer and Water Heater for Any Climate
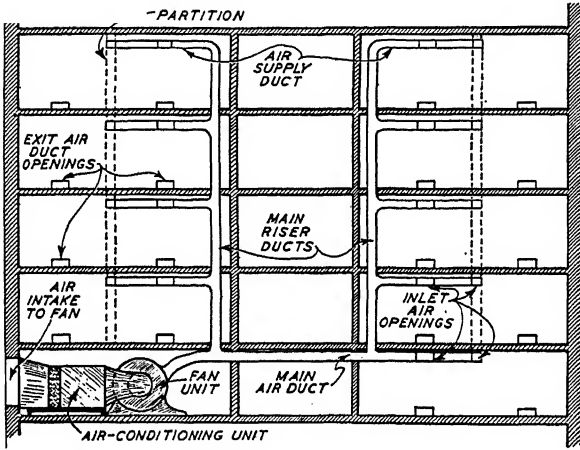
Fig. 44. Air-Conditioning System for Office Building
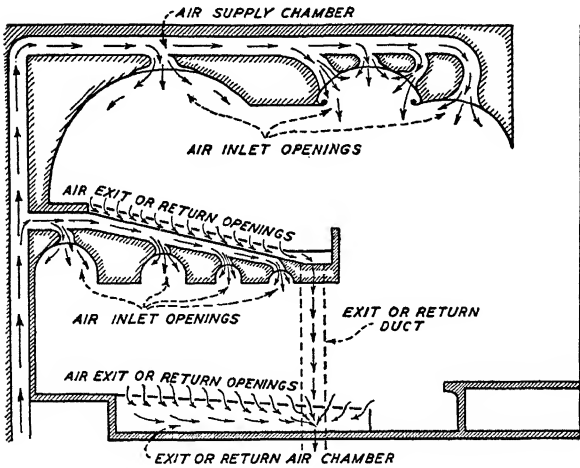
Fig. 45. Air-Conditioning System for Theatre

theatre. Most theatres employ a 100 per cent fan heating and ventilating system with radiators installed only near exits, lobbies, etc. When the weather is not severe, a theatre system is likely to be called upon to do more cooling and ventilating than heating.
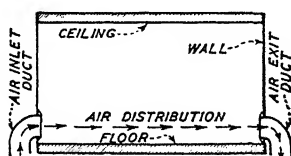
Fig. 46. Openings to Provide Distribution across Lower Section of Room

Methods of Air Distribution. The distribution of air throughout a room or enclosure must be planned to accomplish the desired results depending on requirements. For example, when a split system is used, the air distribution should be as shown in Fig. 47 because the radiator would be under the window. Figs. 46 and 48 are self-explanatory, except as noted by their captions. Incoming air should tend to "push out" the stale air and thus keep up a circulation of fresh, conditioned air.
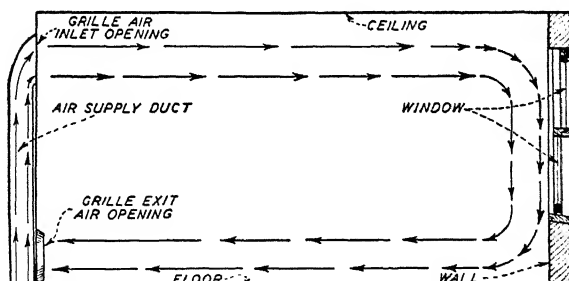
Fig. 47. Openings to Provide Distribution to High Heat Transfer Surfaces

Figs. 49 and 50 show the proper and improper delivery of air from the duct opening to the room. Vanes or blades should always be employed to control the direction and manner of air distribution from duct openings. When low inlet velocities are desired from openings, the duct for some distance back of the opening should be enlarged.

*Design of Fan Systems.   The general procedure for the design of central fan systems is as follows:

(1)   Calculate the heat loss for each room or space to be heated.
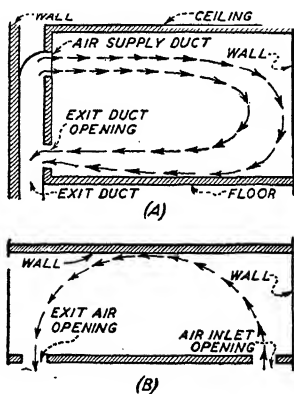
Fig. 48. *A*—Distribution for Interior from Upper Sidewall Openings to Lower Sidewall Openings; *B*—Distribution for Interior from Duct

(2)   Determine volume of outside air to be introduced.

(3)   Assume or calculate temperature of air leaving registers or supply outlets.

(4)   Calculate weight of air to be circulated.

(5)   Estimate temperature loss in duct system.
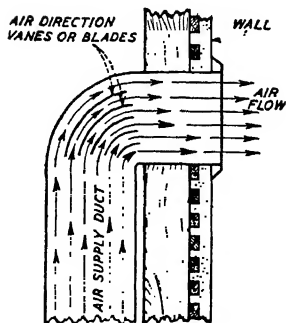
Fig. 49. When Vanes or Blades Are Used in Duct Openings the Air Flow Is Properly Controlled
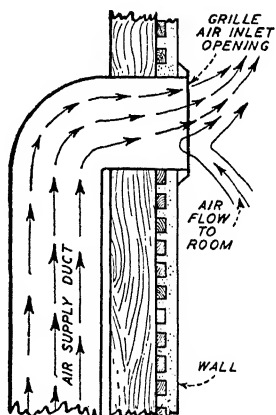
Fig. 50. When no Vanes or Blades Are Used in Duct Openings, the Air Flow Is Erratic and Improper

*Data taken from A.S.H.V.E. Guide 1936, Chapter 22.

(6) Calculate heat to be supplied the heating units and washer.

(7) Select heating units and washer from manufacturers' data and performance curves.

(8) Calculate total heat to be supplied.

(9) Calculate grate area and select boiler. (See Chapter V.)

(10) *Design duct system. (See Chapter XXI.)

(11) Calculate total static pressure of system.

(12) Select fan and motor.

The heat losses should be calculated in accordance with the procedure outlined in the sections on "Transmission Coefficients" and "Heating Loads." If a positive pressure is maintained by the central fan system in the room or space to be ventilated or conditioned, there will ordinarily be very little infiltration of cold outside air through the cracks and crevices of the space. Consequently, the volume of air introduced into the space at the assumed or calculated outlet temperature need only be sufficient to provide for the transmission losses, plus about one-third of the infiltration losses. The exfiltration of heated or conditioned air through the cracks and crevices of the space should be provided for by making the usual allowance for the infiltration losses in arriving at the total heat loss of the space. The air required to make up for this exfiltration of heated or conditioned air will be brought in at the outside air intake and may be included as a part of the outside air necessary for the ventilating requirements. Heat required to raise this air to room temperatures is provided by the coils A, B, C, D, etc., in Figs. 40 to 43 inclusive.

Volume of Outside Air. The volume of outside air should be calculated assuming not less than 10 c.f.m. per person.

The heat required to warm the outside air used for ventilation (H_o) may be determined by means of the following formula:

$$H_o = 0.24(t - t_o)M_o \qquad (15)$$

where 0.24 = specific heat of air at constant pressure.

t = room temperature, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

M_o = weight of outside air to be introduced per hour in pounds = $d_o Q_o$.

Q_o = volume of outside air to be introduced, cubic feet per hour.

d_o = density of air at t_o, pounds per cubic foot.

Example. A building in which the temperature to be maintained is 70°F. requires 10,000 c.f.m. If the outside temperature is 20°F., how much heat will be required to warm the air introduced for ventilation purposes to the room temperature?

Solution. $Q_o = 10,000 \times 60 = 600,000$ c.f.h.; $d_o = 0.08276$ (Table 1, page 15); $M_o = 0.08276 \times 600,000 = 49,656$ lb.; $t = 70°F.$; $t_o = 20°F.$; $H_o = 0.24 \times (70 - 20) \times 49,656 = 595,872$ B.t.u. per hour.

Temperature of Air Leaving Registers. If the system is to function only as a heating system, that is, entirely as a recirculating one, the temperature of the air leaving the register outlets must be assumed. For public buildings, these temperatures may range from 100° to 120° F., whereas for factories and industrial

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*Design of Ducts is beyond the scope of this book. See "Design and Construction of Ducts," published by American Technical Society, Chicago.

buildings the outlet or register temperature may be as high as 140° F. In no case should the outlet temperature exceed these values.

For ventilating or conditioning systems, the temperature of the air leaving the supply outlets may be estimated by means of the following formula:

$$t_y = \frac{H}{60dQ \times 0.24} + 1 \tag{16}$$

where t_y = outlet temperature (F.)

H = heat loss of room to condition, in B.t.u. per hour.

Q = total volume of air to be introduced at the temperature t, cubic feet per minute.

d = density of air, pounds per cubic foot.

If the total temperature t_y as found in Formula (16) is more than 120°F. for public buildings, or 140°F. for factories, etc., these respective outlet temperatures should be used in Formula (17).

$$Q = \frac{H}{60d \times 0.24(t_y - t)} \tag{17}$$

Example. The heat loss for a large enclosure is 100,000 B.t.u. per hour. Also 1,500 c.f.m. are required. The room temperature is 70°F. Determine outlet temperature.

Solution. Substitute in Formula (16).

$$t_y = \frac{100,000}{60 \times 0.07495 \times 1500 \times 0.24} + 70 = 131.7°F.$$

This temperature is excessive; therefore it will be necessary to assume an outlet temperature, taken as 120°F., and to determine the amount of air to be introduced into the room at this temperature to provide for the heat loss. Substitute in Formula (17).

$$Q = \frac{100,000}{60 \times 0.07495 \times 0.24 (120 - 70)} = 1850 \text{ c.f.m.}$$

(at temperature t)

Weight of Air to Be Circulated. The total weight of air to be introduced into the room or space to be heated or conditioned (M) is given by the following formulas:

$$M = \frac{H}{0.24(t_y - t)} = 60dQ \tag{18}$$

$$M = M_o + M_r \tag{19}$$

$$M_o = 60d_oQ_o \tag{20}$$

where d = density of air at temperature t, pounds per cubic foot.

d_o = density of air at temperature t_o, pounds per cubic foot.

Q_o = volume of outside air at temperature t_o.

M_o = weight of outside air, pounds.

M_r = weight of recirculated air, pounds.

Example. Using the data of the preceding example and an outside temperature of 20°F., what will be the values of M, M_o and M_r?

Solution. $d = 0.07495$, $d_o = 0.08276$, $Q = 110,400$, $Q_o = 90,000$, $H = 100,000$.

$$M \frac{100,000}{0.24 \times (120 - 70)} = 8,333 \text{ lbs.}$$

$$M_o = 0.08276 \times 90,000 = 7,448 \text{ lbs.}$$
$$M_r = M - M_o = 8,333 - 7,448 = 885 \text{ lbs.}$$

Temperature Loss in Ducts. The allowances (t_y) to be made for temperature drop through the duct system are as follows:

(1) When the duct system is located in the enclosure to which the air is being delivered, it may be assumed that there is no loss between the reheater coil and the point or points of discharge into the enclosure.

(2) For ducts run underground an allowance shall be made based on the estimated heat loss of the duct, assuming the average temperature of the ground to be 55°F.

(3) For galvanized ducts with the usual ranges of air temperature and velocity, the coefficient of heat transmission may be taken as 1.7 B.t.u. per hour per degree difference between the mean temperature of the air in the duct and that surrounding the duct.

The heat loss may then be expressed by

$$H = 1.7 \, \pi \, DL \left(\frac{t_i + t_y}{2} \right) - t_i \tag{21}$$

and also by

$$H = 0.24 M (t_i - t_y) = 60 \, \frac{\pi}{4} D^2 V d \times 0.24 (t_i - t_y) \tag{22}$$

Equating Formulas (21) and (22)

$$1.7 \, \pi \, DL \left(\frac{t_i + t_y}{2} \right) - t_o = 3.6 \, \pi \, D^2 V d (t_i - t_y)$$

$$\frac{t_i + t_y - 2t_o}{t_i - t_y} = \frac{4.235 D V d}{L} \tag{23}$$

where H = heat loss from the duct, B.t.u. per hour.
   D = diameter of duct, feet.
   L = length of duct, feet.
   t_i = temperature of air entering the duct.
   t_y = temperature of air leaving the duct.
   t_o = temperature of air surrounding the duct.
   M = weight of air passing a given cross section of the duct per hour.
   V = velocity of air in the duct, feet per minute, at specified temperature.
   d = density of the air at the specified temperature at which V is measured.

Usually all the values in Formula (23) may be approximated except t_i, the initial or entering temperature, which can be found readily where the others are known or assumed.

Example 1. Determine the temperature drop in a galvanized duct 20 inches in diameter and 60 feet long carrying air at a velocity of 1200 f.p.m. measured at 70°F., to be delivered at a temperature of 140°F. when the air surrounding the duct is at a temperature of 50° F.

Solution. Substituting in Formula (23),

$$\frac{t_i + 140 - (2 \times 50)}{t_i - 140} = \frac{4.235 \times 1.666 \times 1200 \times 0.07495}{60}$$

$$t_i = 158.7°F$$

$$\text{temperature drop} = 158.7 - 140 = 18.7°F.$$

Example 2. An uninsulated 12 inch diameter galvanized duct extends 50 feet through an unconditioned room to supply 80°F. cool air to an adjacent space. If the average temperature of the air surrounding the duct in the unconditioned room is 100°F. and the velocity of the air through the duct is 1000 f.p.m., measured at 70°F., determine the temperature gain which must be allowed in passing the air through the unconditioned room.

Solution. Substituting in Formula (23),

$$\frac{t_i + 80 - (2 \times 100)}{t_i - 80} = \frac{4.235 \times 1.0 \times 1000 \times 0.07495}{50}$$

$$t_i - 120 = 6.34(t_i - 80)$$
$$-5.34 t_i = 120 - 507$$
$$t_i = 72.3°F.$$

$$\text{temperature gain} = 80 - 72.3 = 7.7°F.$$

Therefore, air having a temperature of 72.3°F. must be introduced at the end of the 50-ft. duct to supply 80°F. air to the conditioned space.

Heat Supplied Heating Units and Washer. In practice the following typical cases may be encountered.

Case A. Building heated 100 per cent by the fan system, air taken from the outside.

Case B. Same as *Case A* except the air is recirculated.

Case C. Part of air recirculated and part taken from the outside.

Case D. Air is delivered to all parts of the building at the same temperature. The relative humidity is kept constant and all air is taken from the outside. (This is not applicable where rooms have individual controls.)

Case E. Outside air, return air, and by-pass air are used with the reheater located in the by-pass air chamber.

In analyzing these cases, the following symbols will be used:

H = heat loss of the room or building, B.t.u. per hour.

H_1 = heat to be supplied to the reheater coil, B.t.u. per hour.

H_2 = heat supplied tempering coil, or tempering coil and preheater, B.t.u. per hour.

H_3 = heat supplied air washer by water heater, B.t.u. per hour.

H_4 = heat to be supplied booster coil, B.t.u. per hour.

M = weight of air to be introduced into the room or building, pounds per hour.

M_r = weight of recirculated air, pounds per hour.

M_b = weight of air by-passing washer, pounds per hour.

M_o = weight of air drawn in from outside, pounds per hour.

t_o = mean temperature of outside air, degrees Fahrenheit.

t = mean air temperature to be maintained in the room or building, degrees Fahrenheit.

t_1 = mean temperature of the air entering the reheater coil.

t_2 = mean temperature of the air leaving the reheater coil.

t_z = temperature loss in the duct system.

t_y = temperature of the air leaving the duct outlets.

t_x = average temperature of air entering tempering coil.

t_w = temperature of air entering washer.

0.24 = specific heat of air at constant pressure.

Examples. *Case A.* In this case $t_x = t_o$.

$$H_2 = 0.24(t_1 - t_o)M_o \qquad (24)$$

$$H_1 = 0.24(t_2 - t_1)M_o \qquad (25)$$

The heat loss H for a certain factory building is 700,000 B.t.u. per hour. The mean inside temperature t to be maintained is 65°F. The assumed outside air temperature t_o is 0°F.; $t_z = 0$, $t_y = t_2$ and is assumed to be 140°F. The temperature leaving the tempering coil is assumed to be 35°F. Required, H_1 and H_2. From Formula (18)

$$M = \frac{700,000}{0.24(140 - 65)} = 38,889 \text{ pounds per hour.}$$

$H_2 = 0.24 \times (35 - 0) \times 38,889 = 326,667$ B.t.u. per hour.

$H_1 = 0.24 \times (140 - 35) \times 38,889 = 980,003$ B.t.u. per hour.

$H_2 + H_1 = 326,667 + 980,003 = 1,306,670$ B.t.u. per hour.

Case B. In this case $t_1 = t$.

$M_r = 38,889$

$M_1 = 0.24(t_2 - t_1)M_r$

$H_1 = 0.24(140 - 65) \times 38,889 = 700,000$ B.t.u. per hour.

Case C. (Fig. 51) A portion of the air circulated is recirculated air and the remainder, as may be required for ventilating purposes, is drawn in from the outside. According to Formulas (18) and (19)

$$M = M_o + M_r = \frac{H}{0.24(t_y - t)}$$

The temperature of the resulting mixture of outside and recirculated air entering the tempering coil is:

$$t_x = \frac{M_o t_o + M_r t}{M} \qquad (26)$$

Assuming that a positive supply of outside air ($d_o = 0.0864$) is required for ventilation at the rate of 90,000 cu. ft. per hour in the preceding example, then $M_o = 0.0864 \times 90,000 = 7776$ lbs. per hour are required, measured at 65°F.

$$M_r = M - M_o = 38,889 - 7776 = 31,113 \text{ lbs.}$$

$$t_x = \frac{7776 \times 0 + 31,113 \times 65}{38,889} = 52°F.$$

$$H_1 = 38,889 \times 0.24(140 - 52) = 821,336 \text{ B.t.u.}$$

This amount of work may be accomplished with one or more banks of heating units, that is, either a single reheater or a tempering coil and reheater.

Case D. (Fig. 52) The maximum relative humidity that may be maintained within the building without the precipitation of moisture on single glazed sash when the outside temperature is 30°F. is approximately 35 per cent. If the inside
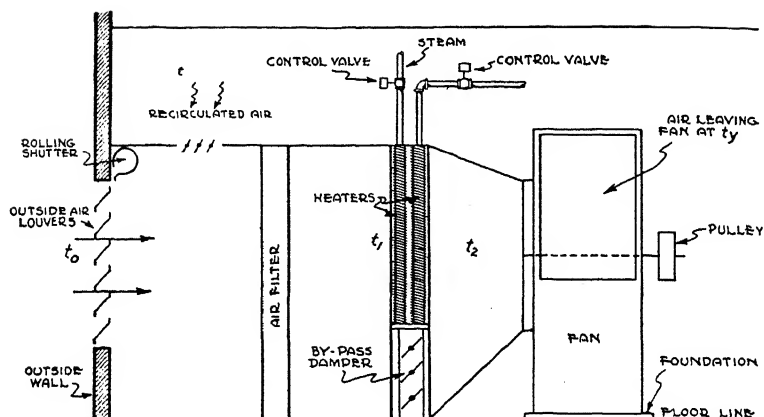
Fig. 51. Combination of Recirculated Air and Outside Air (Case C)

temperature t is 70°F., 35 per cent relative humidity corresponds to a dew-point temperature of 41°F. (See Psychrometric Chart.)

The installation shown in Fig. 52 contemplates the use of a tempering coil, an air washer provided with a water heater, and a reheater. The tempering coil, one section in depth, warms the incoming air to approximately 35°F. to prevent freezing any of the spray water. The air passing through the spray chamber is saturated and leaves at a temperature of $t_1 = 41$°F.

The heat to be supplied the reheater is:

$$H_1 = 0.24(t_2 - 41)M \text{ B.t.u. per hour.}$$

The heat to be supplied the tempering coil is:

$$H_2 = 0.24(35 - t_o)M \text{ B.t.u. per hour.}$$
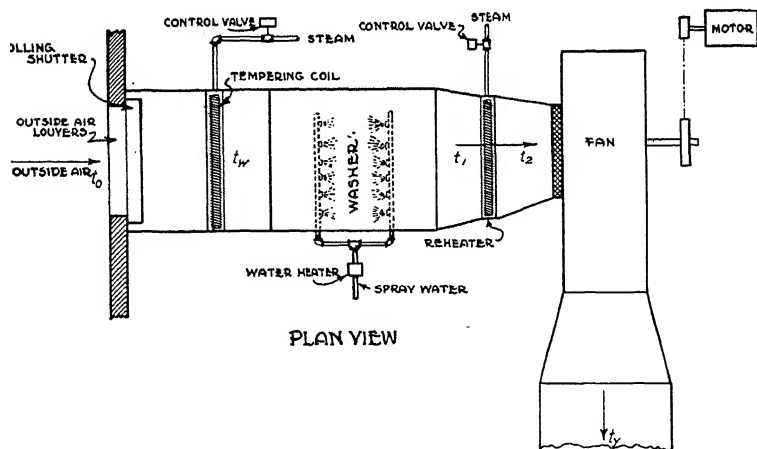
Fig. 52. Outside Air Circulated; Constant Relative Humidity in Room (Case D)

The amount of heat, per pound of air circulated, to be supplied the humidifying washer or humidifier is the difference between the heat content of the assumed dry air entering the washer at a temperature of $t_w = 35°F$. and that of the leaving saturated air at $t_1 = 41°F$. (See Table 300 in Appendix), or:

$$15.657 - 8.397 = 7.26 \text{ B.t.u. per pound of dry air.}$$

The amount of heat required for the washer is:

$$H_3 = 7.26 \ M \text{ B.t.u. per hour.}$$

The total amount of heat required by the apparatus is, therefore:

$$H_1 + H_2 + H_3 \text{ B.t.u. per hour.}$$

If a washer having a *humidifying efficiency* of 67 per cent *without water heater* is employed, it will be necessary to heat the outside air drawn into the apparatus by means of the tempering and preheater coils to such a temperature that the air in passing through the water sprays will become partially saturated (adiabatically) having a moisture content per pound of air equal to saturated air at 41°F. If the incoming air is warmed to $t_w = 88°F$. (requiring a two-section-depth heating unit), it will be cooled in the washer to 64°F. or $88 - 64 = 24$ degrees.

If the *humidifying efficiency* of the washer were 100 per cent, the air would become adiabatically saturated at 52°F. or a temperature drop of $88 - 52 = 30°F$. The efficiency of the washer is, however, only 67 per cent, so that the actual temperature drop will be 0.67×36 degrees or 24 degrees, as used.

The heat to be supplied the reheater is in this case $H_1 = 0.24(t_2 - 64)M$ B.t.u. per hour, and for the tempering coil and preheater is $H_2 = 0.24(88 - t_o)M$. The total heat required is $H_1 + H_2$, no heat being supplied to the washer.

Case E. (Fig. 53). The temperature t_y will ordinarily be different for each room. With H and M fixed, $0.24 \ (t_y - t)M = H$, or

$$t_y = \frac{H}{0.24M} + t$$

In order to provide the proper temperature for each room, a booster coil is generally installed in each supply duct near the outlet to control the outlet temperature t_y. The amount of steam supplied to these booster units is usually controlled automatically by individual thermostats. The heat required by the booster coils depends on the temperature range through which the air is heated and the quantity of air, or

$$H_4 = 0.24(t_y - t_2 - t_z)M \tag{27}$$

Total Heat to Be Supplied. The total heat to be supplied (H') is equal to the sum of the heat requirements of the various heating units and the water heater of the washer, if any, plus the allowance for piping tax, etc. (See preceding *Cases A* to *E*.)

Weight of Condensate. The normal weight of condensate to be handled from central fan systems may be estimated by means of the following formula:

$$W = \frac{60 \times Q \times \Delta t}{55.2 \times h_{fg}} \tag{28}$$

where W = weight of condensate, pounds per hour.

Q = total volume of air, cubic feet per minute.

Δt = temperature rise of air, degrees Fahrenheit.

h_{fg} = latent heat of steam in the system, B.t.u. per pound.

Static Pressure. The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus in-
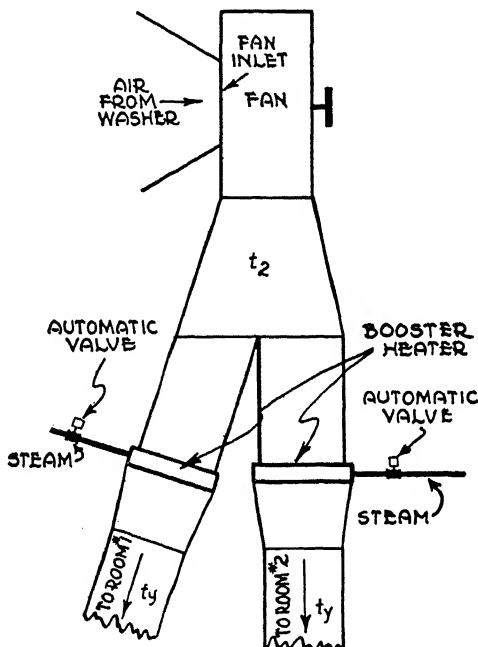

Fig. 53. Outside Air Circulated; Constant Temperature and Relative Humidity Maintained in Each Room (Case E)

volved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed:

(1) Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 inches of water.

(2) A typical oil filter at rated capacity and velocity has a drop of 0.25 in. water.

(3) The loss of one row of a standard make tempering stack equals 0.09 in. water.

(4) The loss of one row of a standard make preheater equals 0.10 in. water.

(5) A standard humidifier at rated velocity may have a loss of about 0.35 in. water.

(6) The loss through one row of a standard make reheater equals 0.12 in. water.

(7) A fair assumption for duct losses on a simple system is 0.25 in. water.

(8) The static pressure for a nozzle type outlet may be taken as 0.1 in. water.

Example 1. A group of three rooms having a total volume of 27,000 cubic feet, a heat loss of 110,000 B.t.u. per hour, and an infiltration loss of 34,200 B.t.u. per hour on the basis of 0°F. outdoors and 70°F. room temperature, is to be heated by a recirculating hot blast heating system with air entering rooms at 116°F. How many cubic feet per minute, measured at 70°F. will be required?

Solution 1.

Substitute in Formula (17)

$$H = 110,000 + 34,200 = 144,300 \text{ B.t.u. per hour.}$$
$$t_y = 116°F.$$
$$t = 70°F.$$
$$Q = \frac{144,300}{60 \times 0.07495 \times 0.24(116 - 70)} = 2,900 \text{ cubic feet per minute.}$$

Example 2. In Example 1, if the hot air loses 4°F. between heater and rooms, how many pounds of steam per hour at 1-pound gauge will the heating sections condense?

Solution 2.

Substitute in Formula (28)

$$Q = 2,900 \text{ c.f.m. (from Example 1)}.$$
$$\Delta t = 116 + 4 - 70 = 50°F.$$
$$h_{fg} = 968 \text{ B.t.u. (from Table 5, page 22, "Fundamentals of Air Conditioning")}$$

$$W = \frac{60 dQ \times 0.24 \times \Delta t}{h_{fg}}$$

$$= \frac{60 \times 0.07495 \times 2900 \times 0.24 \times 50}{968}$$

$$= 161.8 \text{ pounds per hour.}$$

Example 3. The combination hot blast heating and ventilating system for the dining rooms of a hotel is to heat the rooms to 70°F. with 0°F. outside, and permit the exhaust fan from the adjoining kitchen to draw 5,000 c.f.m. from the dining rooms. The transmission losses from the dining rooms total 240,000 B.t.u. per hour. The infiltration into the dining rooms amounts to 1000 c.f.m. from outdoors and 1000 c.f.m. from heater rooms. How many cubic feet per minute, measured at 70°F., must be supplied the dining rooms if the air enters at 112°F.?

Solution 3.

First, find the infiltration loss by substituting in Formula (15)

$t = 70°F.$; $t_o = 0$; $M_o = d \times Q = 0.07495 \times 60 \times 1000 = 4497$ lbs. per hour. In this case d and Q are figured at 70°F. $H_o = 0.24(t - t_o)$; $M_o = 0.24(70 - 0) \times 4497 = 75,550$ B.t.u. per hour.

Next, by substituting in Formula (17) find the cubic feet per hour to be circulated. $H =$ sum of transmission and infiltration losses in room $= 240,000 + 75,550$

$= 315,550$ B.t.u. per hour; $t_y = 112°F.$; $t = 70°F.$; $Q = \dfrac{55.2H}{t - t} \quad \dfrac{55.2 \times 315,550}{112 - 70}$

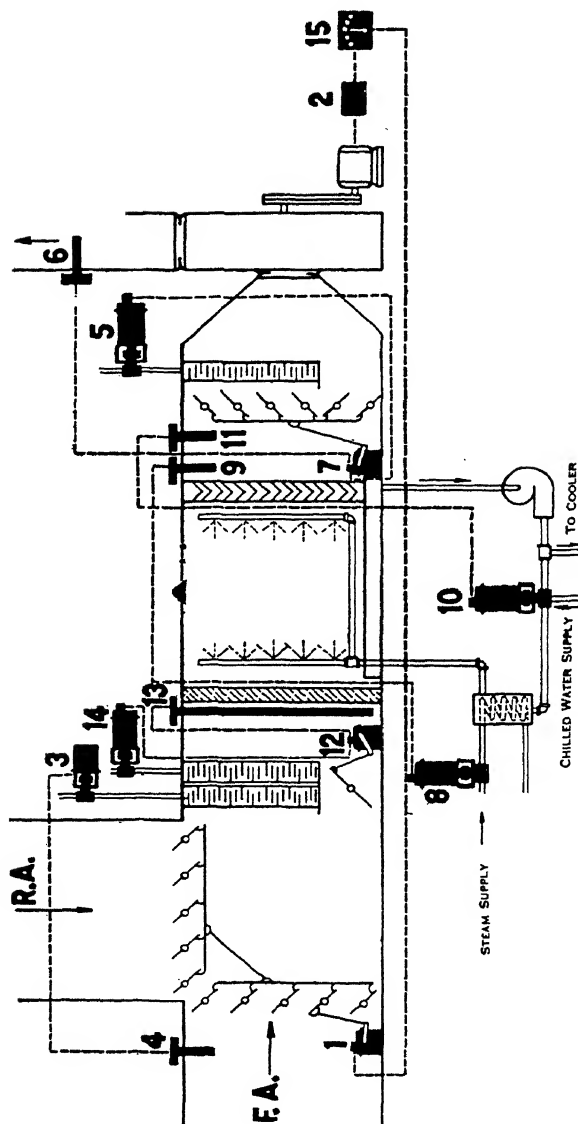$= 414,700$ cu. ft. per hour. c.f.m. $= \dfrac{414,700}{60} = 6912.$

Fig. 54. Control System. This system contains five-position control of fresh air damper, automatic control of by-pass, dampers under preheater and reheater, artificial cooling, and summer and winter humidity control.

Courtesy of Barber Colman Company, Rockford, Ill.

1. Damper control motor	9. Thermostat
2. Relay	10. 3-way valve
3. Positive valve	11. Thermostat
4. Thermostat	12. Damper control motor
5. Throttling valve	13. Thermostat
6. Thermostat	14. Throttling valve
7. Damper control motor	15. 5-point switch
8. Throttling valve	R.A.=Recirculated Air

F.A. = Fresh Air

Central System Controls. Central systems have been explained insofar as general principles and operation are concerned. Such systems must be automatically controlled if they are to be efficient and fill the requirements. The following systems illustrate the automatic control methods which are typical. These illustrations will clearly show the application of dampers, by-pass, etc., as well as controls.

Fig. 54 shows a system where recirculated air use is provided for as well as by-pass around the heaters. The system is for both summer and winter applications. The air washer is that portion of the equipment above the Number 10. Coils and filters are shown in typical symbols as are other main parts. The dampers can be recognized as those symbols consisting of short lines on either side of circles. The explanation of controls for the system shown in Fig. 54 follows.

Temperatures selected for illustration can, of course, be varied, but the same relative setting of the thermostats must be used for proper functioning of the system.

Fresh Air Damper Control. Relay 2 controls damper control motor 1, which in turn operates the fresh air damper in such a manner that when the fan is running the damper is placed in one of its open positions, and when the fan is stopped the damper is closed. Automatic closing of the damper prevents cold air reaching the tempering coils, and possible damage due to freezing when the system is not in operation.

The open position of the fresh air damper is determined by the setting of the 5-point switch. The usual arrangement of these five positions, expressed in per cent of full opening, is 0, 25, 50, 75 and 100. Selection, or the changing, of the position of the fresh air damper makes it possible to regulate the amount of fresh air.

If multi-position control of the fresh air damper is desired, a Microrelay should be added, Microtrol substituted for the 5-position damper control motor, and a rheostat used in place of the 5-point switch. With this combination, the fresh air damper may be positioned exactly in accordance with the position selected by the rheostat.

Recirculated Air Damper Control. The recirculated air damper is connected to the fresh air damper in such a way that when the fresh air damper is fully open the recirculated air damper is fully closed, and vice versa. Intermediate positions are proportional.

Preheater Control. *First Coil.* Thermostat 4 set at 35°F. opens positive valve 3 on the steam line to the preheater whenever the outdoor temperature drops below 35°F., thus avoiding a possibility of freezing in the coils.

Second Coil. Air stream thermostat 13 is installed so that the sensitive element is in contact with the air coming through the by-pass, as well as with that coming through the coils. It is set at 40°F. and operates reversible damper control motor 12, which in turn operates the by-pass damper to maintain a constant temperature of the air entering the air washer. The operation of throttling valve 14 on the steam line is controlled from the auxiliary switches in damper control motor 12. Damper control motor 12 always regulates the temperature as far as possible by the operation of the by-pass damper; valve 14 operates only

when necessary to supply more or less heat after the damper has reached its maximum heating or cooling position.

Upon call for heat, damper control motor *12* starts to close by-pass damper. If the demand for heat continues after damper has reached the maximum heating position (by-pass closed) then throttling valve *14* starts to open, and continues to open until thermostat *13* is satisfied.

Upon call for less heat, throttling valve *14* does not change its position, the cooling being accomplished by damper control motor *12* regulating the by-pass damper. However, when the damper reaches the full cooling position (by-pass open) and there is demand for additional cooling, then throttling valve *14* starts to close, and continues to close until thermostat *13* is satisfied.

The adjustable speed mechanism on both damper control motor *12* and throttling valve *14* makes it possible to regulate the damper and valve, on the job, for a speed best suited to each individual installation.

Humidity Control. *Heating Cycle.* Humidity is maintained in the conditioned space by controlling the dew point temperature of the air leaving the air washer. The air, being saturated at the dew point temperature, will of course have a definite relative humidity when reheated to any higher temperature. Therefore, it is only necessary to control the dew point temperature with thermostat *9* to maintain the desired relative humidity of the air leaving the system. For example, thermostat *9* set at 40°F. will maintain the relative humidity of the air leaving the system at approximately 34%, when reheated to 70°F., or approximately 30% in the conditioned space (allowing 4% for loss). This is a very simple method and is accomplished by operating throttling valve *8* on the steam line to an indirect heater which regulates the temperature of the water to the washer.

Upon call for heat, valve starts to open, and continues opening until thermostat *9* is satisfied, or until the full open position is reached if the demand for heating persists.

Upon call for cooling, the reverse action takes place, valve closing until thermostat *9* is satisfied, or until the full closed position is reached. Valve will stop and remain in any position whenever, and as long as, thermostat is satisfied; it may be restarted in either direction to satisfy demands for heating or cooling.

Cooling Cycle. When the outdoor wet-bulb temperature rises above the point where evaporative cooling from the water in the air washer is not sufficient to hold the dew point temperature at or below 40°F., then thermostat *9* will of course be calling for cooling and keep valve *8* closed. As the dew point temperature rises to 50°F., then thermostat *11* starts to operate three-way valve *10*, admitting the proper proportion of chilled water and recirculated water to maintain a dew point temperature of 50°F. Three-way valve *10* operates in such a manner that on a continued call for heat, the valve admits all recirculated water; and on a continued call for cooling, the valve admits all chilled water. In intermediate positions, accurate proportioning is obtained. The valve operator is of the throttling type, and may be stopped in any position (when demands for heating and cooling are satisfied) and restarted in either direction. Valve *8* will, of course, be closed whenever chilled water is being circulated because the setting of thermostat *9* is lower than the setting of thermostat *11*, and similarly no chilled water will be circulated when valve *8* is open.

In case it is desired to have the differential between thermostat *9* and ther-

mostat *11* very small, valve *10* is prevented from admitting any chilled water when valve *8* is open, and vice versa, by wiring valve *10* through the interlock switches in the valve operator on valve *8*. However, with the average air-conditioning system which is installed for comfort only, the differential between the settings of these thermostats will be at least 10 degrees, as it is desirable to maintain a considerably lower dew point temperature in winter to eliminate excessive window condensation.

Refrigeration Control. The chilled water may be obtained directly from a deep well water supply, ice bunker, or chilled by either mechanical or steam jet refrigeration.

Reheater Control. Thermostat *6* operates reversible damper control motor *7* which in turn operates the face and by-pass damper to maintain a constant temperature of the air leaving the system. Thermostat *6* is set at 70°F. during the heating cycle, and reset to a lower temperature during the cooling cycle. The adjustable speed mechanism on damper control motor *7* makes it possible to regulate the damper, on the job, for a speed best suited for each individual installation.

Fig. 55 shows a year-round air-conditioning system having direct radiation for winter heating, zone or individual room control, and summer and winter humidity control.

The system operates as follows.

Temperatures selected for illustration can, of course, be varied, but the same relative setting of the thermostats must be used for proper functioning of the system.

Fresh Air Damper Control. Relay *2* controls reversing damper control motor *1*, which in turn operates the combination fresh air and recirculated damper in such a manner that when the fan is running the fresh air damper is opened at least to the minimum position and when the fan is stopped the fresh air damper is closed. The minimum position to which the fresh air damper may be opened is usually 25% open, or that point which will admit only sufficient fresh air for ventilation. Automatic closing of the fresh air damper prevents possible damage due to freezing when the system is not in operation. Switch *23* is provided so that the fresh air damper may be closed, even though the fan is running, to permit rapid warm-up after a shutdown period, by the use of full recirculation.

When the temperature of the outside air is above 65°F., it has little or no cooling value; therefore, at 65°F., the setting of thermostat *13*, damper control motor *1* is operated to close fresh air damper to the minimum position.

When the temperature of the outside air is below 65°F., control of damper control motor *1* is automatically shifted to thermostat *14* to proportion automatically the fresh air and recirculated air. Air stream thermostat *14* is set at the lowest temperature desired in the fan discharge, usually about 60°F. It extends across the entire duct and controls from the average temperature in the plenum at this point.

This arrangement of control is most satisfactory, and increases the economy of the system by eliminating the necessity of running the cooling equipment in mild weather whenever, and as long as, the outside air has any value as a cooling medium.

Note: The air stream thermostat is installed so that the sensitive element is in contact with all strata of air. If the arrangement of the duct is such that a thorough mixing of the fresh air and recirculated air is obtained before reaching the thermostat, then a regular insertion type thermostat may be used in place of air stream thermostat *14*.

Preheater Control. Air stream thermostat *4* set at 57°F. operates throttling valve *3* to prevent the average temperature of the air entering the plenum dropping below 57°F.

Note: The air stream thermostat is installed so that the sensitive element is in contact with all strata of air. If the arrangement of the duct is such that a thorough mixing of the fresh air and recirculated air is obtained before reaching the thermostat, then a regular insertion type thermostat may be used in place of air stream thermostat *4*. Because of the fact that the reheater or tempering coil is placed in the recirculated air duct it is possible to use a throttling valve *3* without danger of freezing in the coils.

Humidity Control. *Heating Cycle.* Hygrostat *9* set at 30%, controls the operation of solenoid valve *8* to add moisture and maintain the desired humidity.

Cooling Cycle. Hygrostat *15* set at 55%, controls the operation of relay *10* and will start the compressor at any time the humidity rises above this setting, and keep it running until the humidity is decreased.

Note: Damper control motors *18* also control the operation of the compressor (to take care of the cooling requirements) but hygrostat *15* has controlling preference whenever it is necessary to reduce the relative humidity.

Refrigeration Control. For the sake of simplicity, it is assumed that refrigeration is furnished by a compressor and direct expansion coils. When switch *17* is in the winter position, the cooling equipment is cut out and cannot operate.

Cooling and Dehumidifying. When switch *17* is in the summer position, throttling valve *5* is run to the closed position and remains closed during the entire cooling period. The compressor is controlled through relay *10* by any one of positive damper control motors *18* as described under zone or room control.

By employing an interlocking circuit between relay *10* and reversible damper control motor *1* the compressor can be prevented from starting whenever more than the minimum amount of fresh air is being admitted. This connection is optional, but can be used to advantage where it is known that 65°F. outside air will always give sufficient cooling. If the compressor should be required to operate even though fresh air at 65°F. (or below) is being admitted, then this connection should not be used.

To reduce the relative humidity, hygrostat *15* may operate relay *10* and keep the compressor running even though all damper control motors *18* are in the closed position. When all of the room dampers are closed to the minimum position, the fan is delivering approximately only one-third of its capacity, due to the action of the static pressure regulator. Therefore, due to the lower velocity, the air is in contact with the cooling coils for a greater length of time, which provides more effective dehumidification. The result of discharging this small volume of dehumidified air into the rooms, is to lower the humidity without lowering the dry-bulb temperature appreciably.

Reheater Control. When switch *17* is in the summer position, throttling valve *5* is run to the closed position, and remains closed, regardless of the position of the face and by-pass dampers.

Heating Cycle. When switch *17* is in the winter position, the cooling equipment is cut out and cannot operate. Thermostat *6*, set at 70°F., operates reversible damper control motor *7*, which in turn operates the face and by-pass damper to maintain the proper temperature of the air leaving the system. This is accomplished by regulating the amount of air by-passed under the coil and

Fig. 55. Control System. This system has zone or individual room control, direct radiation for winter heating, and summer and winter humidity control. It is a year-round system.

Courtesy of Barber Colman Company, Rockford, Ill.
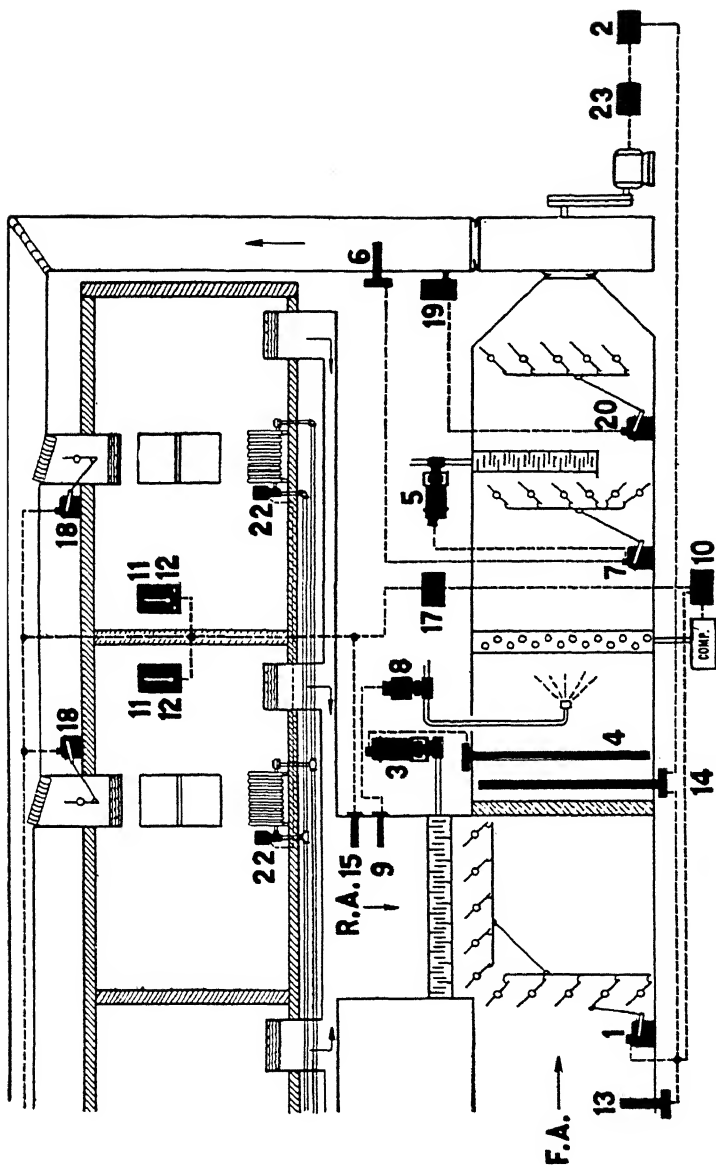
1. Damper control motor
2. Relay
3. Throttling valve
4. Thermostat
5. Throttling valve
6. Thermostat
7. Damper control motor
8. Water valve
9. Hygrostat
10. Relay

} Thermostats

Thermostat
Thermostat
Hygrostat
Double throw switch
Damper control motor
Static pressure regulator
Damper control
Positive valve

23. Switch

that passed through the coil. Upon continued demands for heating, all air is directed through the coil; upon continued demands for cooling, all air is directed under the coil through the by-pass. In intermediate positions accurate proportioning is obtained. Throttling valve 5 on the steam line is controlled from the auxiliary switches in damper control motor 7. The adjustable speed mechanism on both damper control motor 7 and throttling valve 5 makes it possible to regulate the dampers and valve, on the job, for a speed suited to each installation.

Static Pressure Control. Static pressure regulator 19 controls reversing damper control motor 20 which in turn operates the damper to regulate the amount of air entering the fan. In this way, any desired constant pressure is maintained in the discharge duct at all times, regardless of the number of zone or room dampers that are open.

Zone or Room Control. *Cooling Cycle.* Each thermostat 11 controls its respective positive damper control motor 18 which in turn operates the shut-off damper to the zone or room. These dampers operate so that they are always in either the full open or closed position. In the closed position, the dampers will still admit air equal to about one-third of that admitted in the full open position, or that amount required for ventilation. On call for cooling, dampers open; on call for heat, dampers close.

When all dampers are closed, the compressor will stop; any one damper opening will cause the compressor to start and continue running as long as one or more dampers are open.

Heating Cycle. When switch 17 is turned to the winter position, all damper control motors 18 are run to the full open position, the dampers remaining open during the entire heating cycle. Each thermostat 12 controls its respective positive zone or radiator valve 22 in such a manner that on a call for heat, valves open; and on a call for cooling, valves close.

Fig. 56 shows a typical illustration of a control system for direct expansion cooling using two compressors.

For purpose of illustration we shall describe this control system using 10- and 20-ton compressors.

Temperature Control. Microtherm 1 controls the operation of program switch 3, which in turn regulates the operation of the compressors to maintain the desired room temperature.

On demand for cooling, microtherm 1 starts program switch 3 which in turn starts the 10-ton compressor. If the demand continues, the 10-ton compressor is cut off and the 20-ton compressor started, with solenoid valve 2 in the closed position, giving 15-ton capacity. Upon further demand for cooling, solenoid valve 2 opens, giving 20-ton capacity. If the demand continues, solenoid valve 2 is closed and the 10-ton compressor restarted, giving 25-ton capacity. Finally, if the demand still continues, solenoid valve 2 reopens, giving 30-ton capacity. On call for less cooling, the reverse action takes place.

The following shows the sequence of operation:

Steps	Capacity Available	10-Ton Compressor	20-Ton Compressor	Solenoid Valve (2)
1st	10 Tons	On	Off	Closed
2nd	15 Tons	Off	On	Closed
3rd	20 Tons	Off	On	Open
4th	25 Tons	On	On	Closed
5th	30 Tons	On	On	Open

Humidity Control. Hygrostat *4* also controls the operation of program switch *3* and is so wired that it has controlling preference over microtherm *1* and will keep the compressors running whenever, and as long as, necessary to maintain the desired relative humidity.
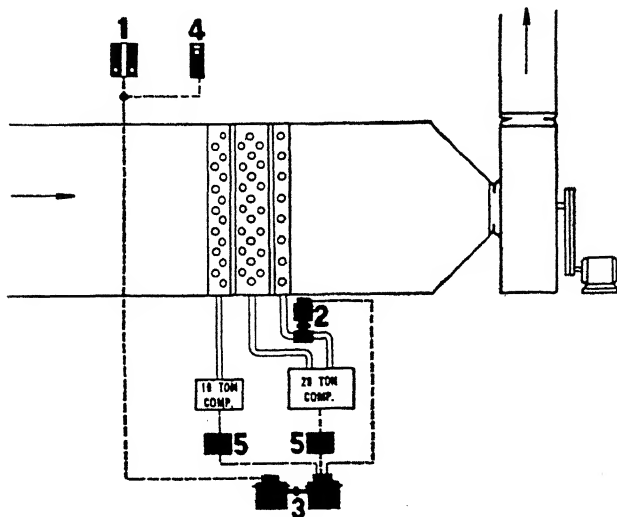

Fig. 56. Control System Illustrating Two Compressors of Different Sizes Controlled in Such a Manner that Five Different Capacities are Available

1. Microtherm 3. Control switch
2. Solenoid refrigerant valve 4. Hygrostat
5. Relays

Courtesy of Barber Colman Company, Rockford, Ill.

Compressor Control. The 20-ton compressor is connected to two separate sets of coils, one of approximately 15-ton capacity, and the other 5-ton capacity. Solenoid valve *2*, installed on the liquid line to the smaller coil, cuts off approximately 5-ton capacity when it is closed. When it is open, full capacity of the 20-ton compressor is permitted.

Program switch *3* is driven by a reversing motor which may be stopped in any position whenever microtherm *1* is satisfied, and restarted in either direction to meet demands for more or less cooling. In the "off" position both compressors are stopped. The motor drives a series of cam-operated switches to start or stop the compressors, or open or close solenoid valve *2*, in the desired sequence with predetermined intervals. An adjustable speed mechanism makes it possible to regulate the program switch so that its speed of operation will be best suited to each individual installation.

This system permits the greatest flexibility with maximum economy and the minimum of equipment.

Fig. 57 shows a typical illustration of chilled water control to supply one surface cooler (coil) or air washer.

Compressor Control. Thermostat *1* set at 45°F. controls the operation

of the compressor through relay *3* to maintain a constant temperature of the chilled water leaving the cooler. On call for cooling, compressor starts and continues to run until thermostat *1* is satisfied. Thermostat *2* set at 35°F. is wired
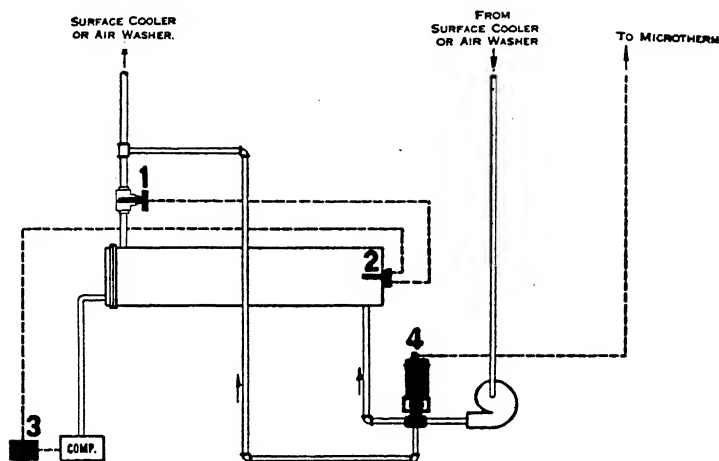

Fig. 57. Typical Control System for an Air-Conditioning System Using Either Cooling Coils or an Air Washer

1. Immersion thermostat   3. Relay
2. Immersion thermostat   4. 3-way valve
Courtesy of Barber Colman Company, Rockford, Ill.

so it will operate relay *3*, stopping the compressor if the temperature of the water in the cooler should drop to 35°F. This precaution is taken to guard against damage due to freezing if the pump should stop; in which case thermostat *1* would not be effective because of its location in the circulating line from the cooler.

The motor starter furnished with the compressor motor must of course be used in addition to relay *3*.

Control of Surface Cooler or Air Washer. Three-way valve *4* controlled by either a room or duct type microtherm, regulates the proper proportion of chilled water and recirculated water to maintain the desired temperature. It operates in such a manner that on a continued call for heat the valve admits all recirculated water, and on a continued call for cooling it admits all chilled water. In intermediate positions accurate proportioning is obtained. The valve operator is of the throttling type, and may be stopped in any position (when demands for heating and cooling are satisfied) and restarted in either direction.

Fig. 58 shows a layout illustrating the control of a typical cabinet type air conditioner.

Cooling Cycle. Thermostat *1* controls the operation of the compressor. On call for cooling, the compressor starts and continues to run until thermostat is satisfied.

Heating Cycle. Thermostat *2* controls the operation of positive valve *5*. On call for heat, valve opens; on call for cooling, valve closes.

Fan Control. The fan may be wired so it will run whenever either the compressor is running or valve *5* is open. A manually operated switch may be provided to allow the fan to run continuously when air circulation is desired.
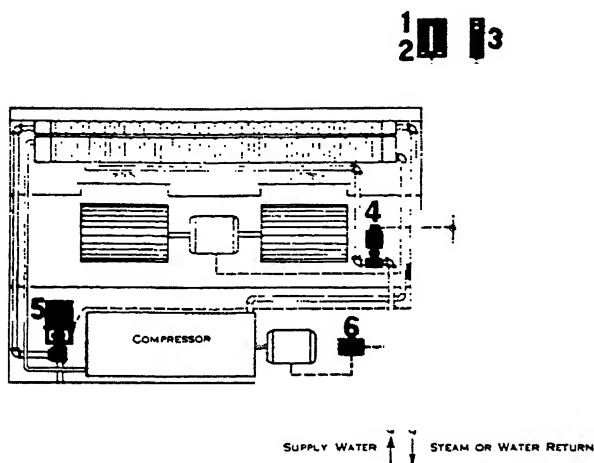

Fig. 58. Control System for a Year-Round Cabinet Air Conditioner

1 and 2. Summer and winter thermostat	4. Solenoid water valve
	5. Positive valve
3. Hygrostat	6. Relay

Courtesy of Barber Colman Company, Rockford, Ill.

Humidity Control. *Heating Cycle.* Hygrostat *3* controls the operation of solenoid valve *4* to add moisture and maintain the desired relative humidity.

If desired, thermostats *1* and *2* may be furnished with their adjusting levers mechanically connected to prevent any possibility of valve *5* being open when the compressor is running, and vice versa.
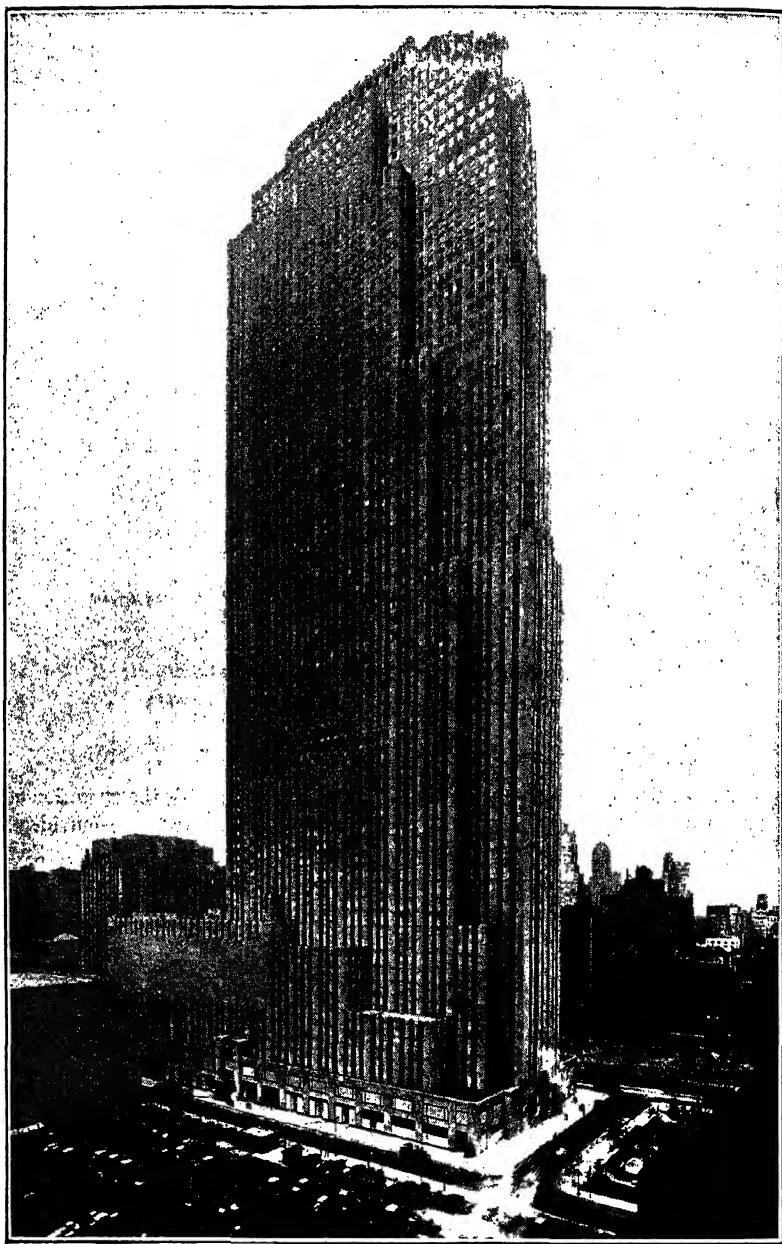
RADIO CITY BUILDING, NEW YORK CITY
This Is Typical of New Buildings Being Air Conditioned
Photo by Underwood & Underwood

CHAPTER VII

RADIATORS

A steam or hot-water heating plant consists of a boiler, of pipes for conveying steam or hot water, and of radiators. This chapter is devoted to radiators and to the selection of radiators, which are used in all types of heating plants.

Types and Location of Radiators. The five common types of radiators are direct, suspended, built-in, and wall radiators, and branch and miter coils.

Direct Radiators. The general forms of these cast-iron sectional radiators are shown in Figs. 59 and 60. Radiators of this type are made in as many sections as needed to provide heating surface of a specified amount. The sections are connected at the bottom by special nipples. Steam entering at one end fills the bottom of the radiator
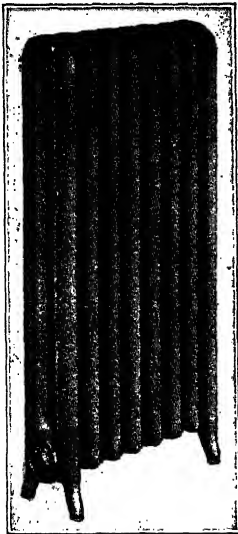

Fig. 59. Peerless Two-Column
Cast-Iron Radiator
(Rating Tables can be secured
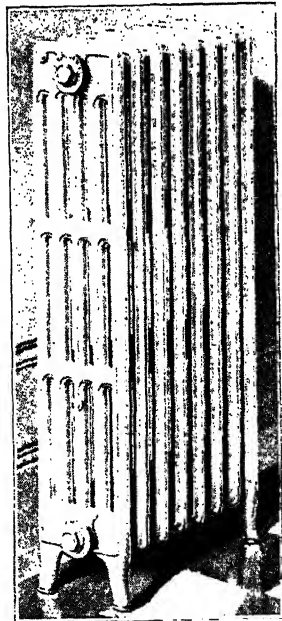from manufacturer)

Fig. 60. Corto Tube Type Cast-
Iron Radiator

Courtesy of American Radiator Company, Chicago

and, being lighter than air, rises through the loops and forces the air downward and toward the other end.   There it is discharged through an air-valve placed about midway of the last section.   For one-pipe steam work the supply-leg section is constructed with a
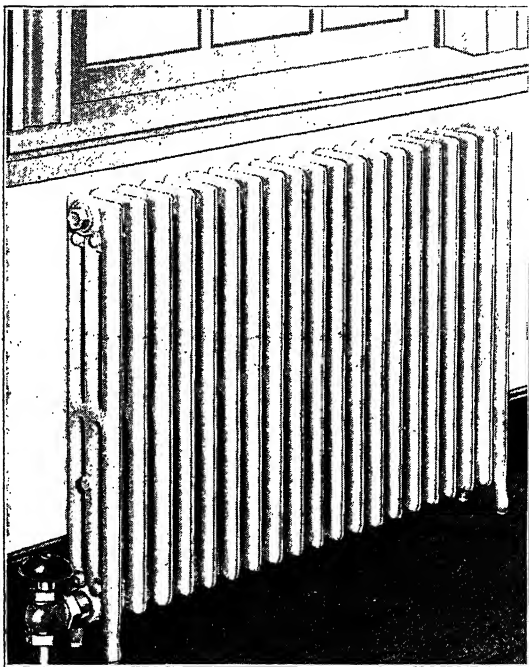

Fig. 61. Arco Tube Type Cast-Iron Radiator
Courtesy of American Radiator Company, Chicago

low-drip hub, and for two-pipe steam work the return-leg section is constructed with a low-drip hub.

Old and New Types.   During the last few years the design of radiators has changed from the obsolete column radiator, Fig. 59, to the new types, Figs. 60, 61, and 62.   The latter are more efficient, require less space, and present a better appearance.   The type shown in Fig. 59 probably will be in use for many years in older buildings but the radiators shown in Figs. 60, 61, and 62 are types for modern buildings.

Radiators like those shown in Figs. 59, 60, and 61 generally are set up on the floor and placed under or near a window—which is the

coldest place in the enclosure during the heating season. They should be at least 2½ inches from the wall to allow a free circulation of air behind them.

Suspended Radiators. The suspended type of radiator, Fig. 62, is designed to be hung below the window sill. The sill is generally of
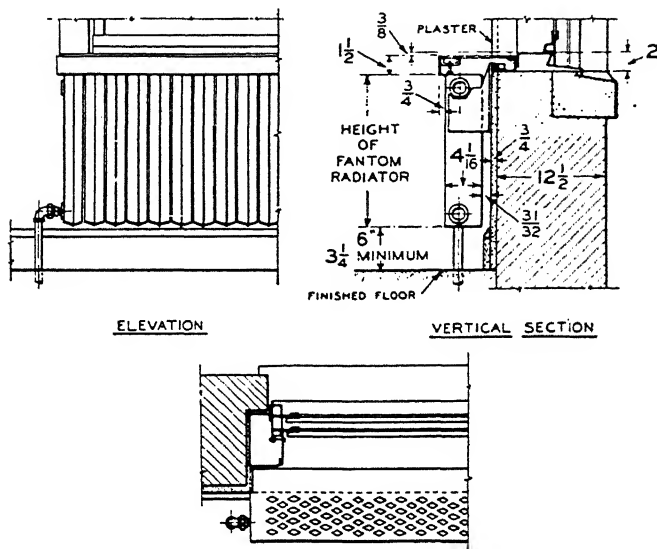

ELEVATION   VERTICAL SECTION

Fig. 62. Suspended Type Cast-Iron Fantom Window Radiator
Courtesy of American Radiator Company, Chicago

open design, as shown in the illustration, to allow the upward movement of heat. This radiator requires little space and acts on the flue principle. Fig. 62 shows a typical installation with dimensions.

Built-in Convectors. This type, Fig. 63, a complete enclosure for general use, can be free standing or partially recessed and has a removable front. The cabinet type convector is shown in Fig. 64.

Wall Radiators. Wall or ceiling type radiators, Fig. 65, can be hung in multiple units along a wall or suspended from a ceiling. They require little space and are off the floor. They are useful especially in basement areas where radiators must be above the boiler level to allow the proper functioning of the circulation system.

Branch and Miter Coils. These coils, Fig. 66 and 67, are generally made of 1- or 1¼-inch wrought-iron pipe. They can be hung on the

LINING LENGTH = CONV. LENGTH PLUS 1½"
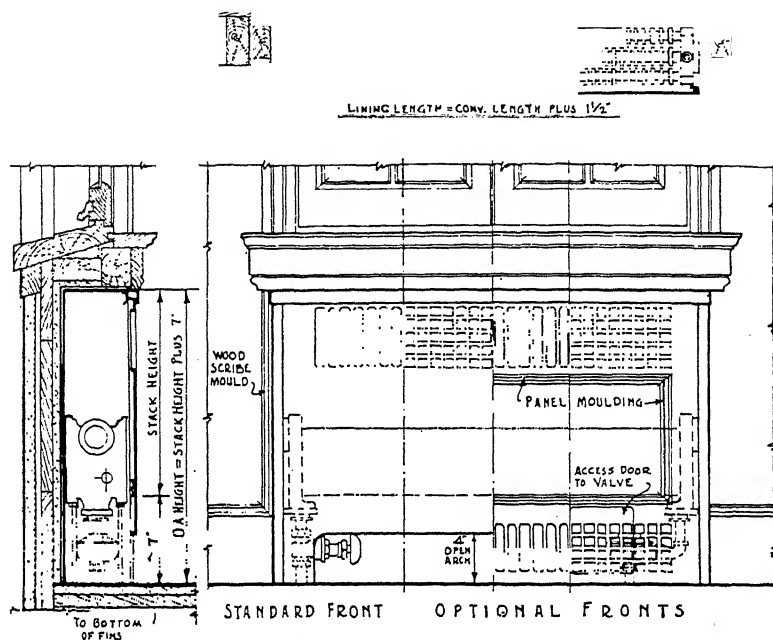
STANDARD FRONT   OPTIONAL FRONTS

Fig. 63. Built-In Convector
Courtesy of American Radiator Company, Chicago

walls of an enclosure by means of hooks, or they can be suspended from overhead on hangers. The coils are used usually in factory or similar heating installations.

Method A—Estimating Radiator Requirements. A standard procedure of heat loss calculation is helpful in practical field work. Where the type of construction is known definitely, this standard procedure can be used to ascertain not only the radiation requirements but also the actual size and kind of radiator to use. This standard procedure is only an estimate but is accurate enough for practical and general use. To explain the principle and application of this standard procedure, the following typical example is used:

*Example 1. Determine the square feet of steam radiation required to maintain 80°F. temperature in a room 15'0"×14'0"×8'0" ceiling, north wall exposed, having two 3'0"×5'0" windows with storm sash, and a 4'0"×5'0" skylight in ceiling having single glass, when it is −20°F. outside temperature. (See Fig. 68.)

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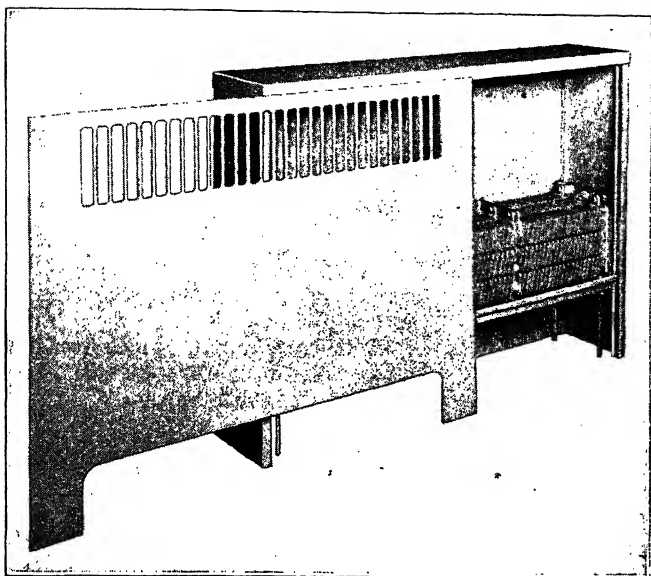*Data Courtesy of American Radiator Company, New York.

Fig. 64. Cabinet Type Convector
(Rating Tables can be secured from manufacturer)
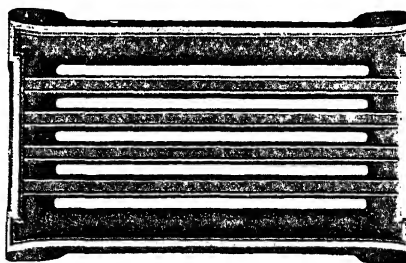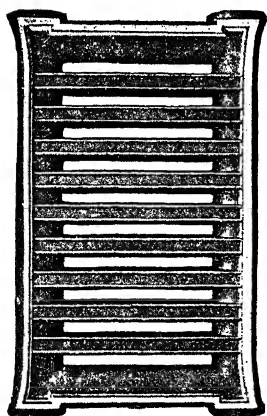*Courtesy of American Radiator Company, Chicago*

Fig. 65. Rococo Wall Radiators
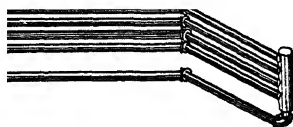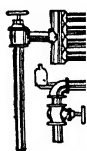*Courtesy of American Radiator Company, Chicago*

Fig. 66. Common Form of "Branch" Coil for Circulation of Direct Steam

Fig. 67. Miter Coil

(1) **Construction Details:**

(a) Exposed Wall...Shingles, paper, sheathing, studding, ½-inch insulation, plaster.

(b) Flat Roof......Paper, 1½-inch boards, ½-inch insulation, joists, lath, and plaster. A 4′0″×5′0″ skylight.

(c) Windows........Two windows—storm sash.

(d) Assume a fireplace.

(2) *Coefficients (U). In this example the U values are as follows:

Exposed wall....................................$U=$ .19
Flat roof.......................................$U=$ .19
Glass...$U=1.13$

It is considered good practice to assume one additional air change per hour, because of the fireplace.

(3) **Calculation of Areas and Volume.** As far as areas are concerned, only the exposed wall, roof, and windows are taken into consideration, assuming that all other walls are adjacent to heated rooms.
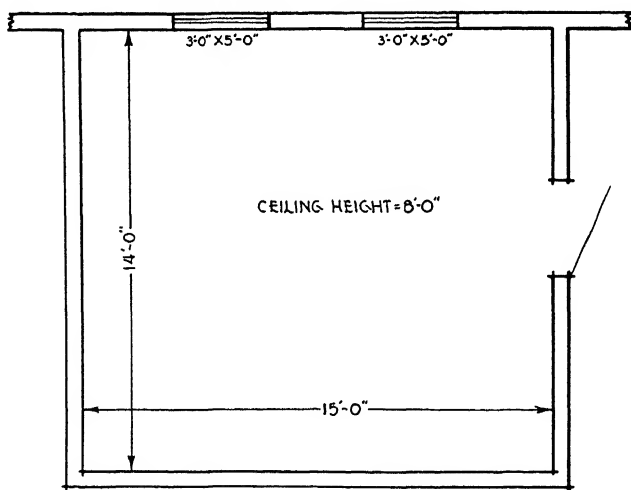

Fig. 68. Floor Plan

*See Tables 7, 8, and 9, Chapter II.

North Wall:

 Wall $15' \times 8'$ $= 120$ square feet
 Glass $3' \times 5' \times 2 = 30$ square feet
    Net Wall $= 90$ square feet

Roof:

 Ceiling $14' \times 15' = 210$ square feet
 Glass skylight
    $4' \times 5' = 20$ square feet
 Net ceiling $= 190$ square feet

Volume:

 $14' \times 15' \times 8'$ $= 1{,}680$ cubic feet

(4) With the values determined in step 3, refer to Table 32. For the net wall area of 90 square feet, under the column headed $U = .19$ (or the nearest thereto), find a number nearest 90 square feet. In this case, it is 85 square feet. At the extreme left of the table is the amount of radiation required, namely 5 square feet (240 B.t.u. per square foot).

To find the amount of radiation for 30 square feet of glass, find number 30 in the column headed "Sq. Ft. Glass" (Table 32) and read 10 square feet radiation in left-hand column. As storm sash is used, divide this result by 2.

Proceed in a similar manner for all other items. The following tabulation may then be set up.

North Wall:

 Wall (90 square feet) requires 5 square feet radiation (240 B.t.u. per
  square foot)
 Glass (30 square feet) requires 10 square feet radiation for single windows
 For storm sash $\dfrac{10}{2}$ requires 5 square feet (240 B.t.u. per square foot).

Roof:

 Ceiling (190 square feet) requires 11 square feet (240 B.t.u. per square foot)
 Glass skylight (20 square feet) requires 7 square feet (240 B.t.u. per square
  foot)
 One air change requires 9 square feet. Two air changes thereafter—18
  square feet (240 B.t.u. per square foot).

(5) While the coefficients of transmission allow for an average wind velocity of 15 miles per hour, further allowance must be made for direction of the prevailing wind in any given locality. This is done by adding 15% to the wall and glass transmission losses and infiltration losses on the sides of the building exposed to or nearly facing the direction of the prevailing winds. In this example assume wind direction to be north—the allowance is then made as follows:

North Wall:

 Wall.................5 square feet $\times$ 1.15* = 5.75 square feet
 Glass.................5 square feet $\times$ 1.15 = 5.75 square feet

*1.15 equals 1+15%, or the method used to increase an area by 15%.

Table 32. Steam—0°F.=70°F.—240 B.t.u. per Square Foot

Sq. Ft. Radiation	Cubic Content	*Sq. Ft. Glass, Single	U Values													
			.30	.28	.26	.24	.22	.20	.18	.16	.14	.12	.10	.08	.06	.04
1	190	3	11	12	13	14	16	17	19	21	24	29	34	43	57	86
2	380	6	22	24	26	28	32	34	38	42	48	58	68	86	114	172
3	570	9	33	36	39	42	48	51	57	63	72	87	102	129	171	258
4	760	12	44	48	52	56	64	68	76	84	96	116	136	172	228	344
5	950	15	55	60	65	70	80	85	95	105	120	145	170	215	285	430
6	1140	18	66	72	78	84	96	102	114	126	144	174	204	258	342	516
7	1330	21	77	84	91	98	112	119	133	147	168	203	238	301	399	602
8	1520	24	88	96	104	112	128	136	152	168	192	232	272	344	456	688
9	1710	27	99	108	117	126	144	153	171	189	216	261	306	387	513	774
10	1900	30	110	120	130	140	160	170	190	210	240	290	340	430	570	860
11	2090	33	121	132	143	154	176	187	209	231	264	319	374	473	627	946
12	2280	36	132	144	156	168	192	204	228	252	288	348	408	516	684	1032
13	2470	39	143	156	169	182	208	221	247	273	312	377	442	559	741	1118
14	2660	42	154	168	182	196	224	238	266	294	336	406	476	602	798	1204
15	2850	45	165	180	195	210	240	255	285	315	360	435	510	645	855	1290
16	3040	48	176	192	208	224	256	272	304	336	384	464	544	688	912	1376
17	3230	51	187	204	221	238	272	289	323	357	408	493	578	731	969	1462
18	3420	54	198	216	234	252	288	306	342	378	432	522	612	774	1026	1548
19	3610	57	209	228	247	266	304	323	361	399	456	551	646	817	1083	1634
20	3800	60	220	240	260	280	320	340	380	420	480	580	680	860	1140	1720
21	3990	63	231	252	273	294	336	357	399	441	504	609	714	903	1197	1806
22	4180	66	242	264	286	308	352	374	418	462	528	638	748	946	1254	1892
23	4370	69	253	276	299	322	368	391	437	483	552	667	782	989	1311	1978
24	4560	72	264	288	312	336	384	408	456	504	576	696	816	1032	1368	2064
25	4750	75	275	300	325	350	400	425	475	525	600	725	850	1075	1425	2150
26	4940	78	286	312	338	364	416	442	494	546	624	754	884	1118	1482	2236
27	5130	81	297	324	351	378	432	459	513	567	648	783	918	1161	1539	2322
28	5320	84	308	336	364	392	448	476	532	588	672	812	952	1204	1596	2408
29	5510	87	319	348	377	406	464	493	551	609	696	841	986	1247	1653	2494
30	5700	90	330	360	390	420	480	510	570	630	720	870	1020	1290	1710	2580
31	5890	93	341	372	403	434	496	527	589	651	744	899	1054	1333	1767	2666
32	6080	96	352	384	416	448	512	544	608	672	768	928	1088	1376	1824	2752
33	6270	99	363	396	429	462	528	561	627	693	792	957	1122	1419	1881	2838
34	6460	102	374	408	442	476	544	578	646	714	816	986	1156	1462	1938	2924
35	6650	105	385	420	455	490	560	595	665	735	840	1015	1190	1505	1995	3010
36	6840	108	396	432	468	504	576	612	684	756	864	1044	1224	1548	2052	3096
37	7030	111	407	444	481	518	592	629	703	777	888	1073	1258	1591	2109	3182
38	7220	114	418	456	494	532	608	646	722	798	912	1102	1292	1634	2166	3268
39	7410	117	429	468	507	546	624	663	741	819	936	1131	1326	1677	2223	3354
40	7600	120	440	480	520	560	640	680	760	840	960	1160	1360	1720	2280	3440
41	7790	123	451	492	533	574	656	697	779	861	984	1189	1394	1763	2337	3526
42	7980	126	462	504	546	588	672	714	798	882	1008	1218	1428	1806	2394	3612
43	8170	129	473	516	559	602	688	731	817	903	1032	1247	1462	1849	2451	3698
44	8360	132	484	528	572	616	704	748	836	924	1056	1276	1496	1892	2508	3784
45	8550	135	495	540	585	630	720	765	855	945	1080	1305	1530	1935	2565	3870
46	8740	138	506	552	598	644	736	782	874	966	1104	1334	1564	1978	2622	3956
47	8930	141	517	564	611	658	752	799	893	987	1128	1363	1598	2021	2679	4042
48	9120	144	528	576	624	672	768	816	912	1008	1152	1392	1632	2064	2736	4128
49	9310	147	539	588	637	686	784	833	931	1029	1176	1421	1666	2107	2793	4214
50	9500	150	550	600	650	700	800	850	950	1050	1200	1450	1700	2150	2850	4300

*For storm sash use one-half of these values.

Allowance for exposure: Add 15% to wall and glass transmission losses also to infiltration l for wall or walls facing or adjacent to direction of prevailing winds.

If the U values selected are not listed above, the table should be used as follows: Ex: U =.60. Use column U =.30 and multiply results by 2.

To correct for temperatures other than 0°F.-70°F. (see Example 1).

Table 33. Hot Water—0°F.-70°F.—150 B.t.u. per Square Foot
170°F. Average Water Temperature

Sq. Ft. Radiation	Cubic Content	*Sq. Ft. Glass, Single	U Values													
			.30	.28	.26	.24	.22	.20	.18	.16	.14	.12	.10	.08	.06	.04
1	118	1.9	7	8	8.5	9	10	10.5	12	13	15	18	21	27	36	54
2	236	3.8	14	16	17	18	20	21	24	24	26	30	36	42	54	72
3	354	5.7	21	24	25	27	30	31	36	36	39	45	54	63	81	108
4	472	7.6	28	32	34	36	40	42	48	48	52	60	72	84	108	144
5	590	9.5	35	40	42	45	50	52	60	60	65	75	90	105	135	180
6	708	11.4	42	48	51	54	60	63	72	78	90	108	126	162	216	324
7	826	13.3	49	56	59	63	70	73	84	91	105	126	147	189	252	378
8	944	15.2	56	64	68	72	80	84	96	104	120	144	168	216	288	432
9	1062	17.1	63	72	76	81	90	94	108	117	135	162	189	243	324	486
10	1180	19.0	70	80	85	90	100	105	120	130	150	180	210	270	360	540
11	1298	20.9	77	88	93	99	110	115	132	143	165	198	231	297	396	594
12	1416	22.8	84	96	102	108	120	126	144	156	180	216	252	324	432	648
13	1534	24.7	91	104	110	117	130	136	156	169	195	234	273	351	468	702
14	1652	26.6	98	112	119	126	140	147	168	182	210	252	294	378	504	756
15	1770	28.5	105	120	127	135	150	157	180	195	225	270	315	405	540	810
16	1888	30.4	112	128	136	144	160	168	192	208	240	288	336	432	576	864
17	2006	32.3	119	136	144	153	170	178	204	221	255	306	357	459	612	918
18	2124	34.2	126	144	153	162	180	189	216	234	270	324	378	486	648	972
19	2242	36.1	133	152	161	171	190	199	228	247	285	342	399	513	684	1026
20	2360	38.0	140	160	170	180	200	210	240	260	300	360	420	540	720	1080
21	2478	39.9	147	168	178	189	210	220	252	273	315	378	441	567	756	1134
22	2596	41.8	154	176	187	198	220	231	264	286	330	396	462	594	792	1188
23	2714	43.7	161	184	195	207	230	241	276	299	345	414	483	621	828	1242
24	2832	45.6	168	192	204	216	240	252	288	312	360	432	504	648	864	1296
25	2950	47.5	175	200	212	225	250	262	300	325	375	450	525	675	900	1350
26	3068	49.4	182	208	221	234	260	273	312	338	390	468	546	702	936	1404
27	3186	51.3	189	216	229	243	270	283	324	351	405	486	567	729	972	1458
28	3304	53.2	196	224	238	252	280	294	336	364	420	504	588	756	1008	1512
29	3422	55.1	203	232	246	261	290	304	348	377	435	522	609	783	1044	1566
30	3540	57.0	210	240	255	270	300	315	360	390	450	540	630	810	1080	1620
31	3658	58.9	217	248	263	279	310	325	372	403	465	558	651	837	1116	1674
32	3776	60.8	224	256	272	288	320	336	384	416	480	576	672	864	1152	1728
33	3894	62.7	231	264	280	297	330	346	396	429	495	594	693	891	1188	1782
34	4012	64.6	238	272	289	306	340	357	408	442	510	612	714	918	1224	1836
35	4130	66.5	245	280	297	315	350	367	420	455	525	630	735	945	1260	1890
36	4248	68.4	252	288	306	324	360	378	432	468	540	648	756	972	1296	1944
37	4366	70.3	259	296	314	333	370	388	444	481	555	666	777	999	1332	1998
38	4484	72.2	266	304	323	342	380	399	456	494	570	684	798	1026	1368	2052
39	4602	74.1	273	312	331	351	390	409	468	507	585	702	819	1053	1404	2106
40	4720	76.0	280	320	340	360	400	420	480	520	600	720	840	1080	1440	2160
41	4838	77.9	287	328	348	369	410	430	492	533	615	738	861	1107	1476	2214
42	4956	79.8	294	336	357	378	420	441	504	546	630	756	882	1134	1512	2268
43	5074	81.7	301	344	365	387	430	451	516	559	645	774	903	1161	1548	2322
44	5192	83.6	308	352	374	396	440	462	528	572	660	792	924	1188	1584	2376
45	5310	85.5	315	360	382	405	450	472	540	585	675	810	945	1215	1620	2430
46	5428	87.4	322	368	391	414	460	483	552	598	690	828	966	1242	1656	2484
47	5546	89.3	329	376	399	423	470	493	564	611	705	846	987	1269	1692	2538
48	5664	91.2	336	384	408	432	480	504	576	624	720	864	1008	1296	1728	2592
49	5782	93.1	343	392	416	441	490	514	588	637	735	882	1029	1323	1764	2646
50	5900	95.0	350	400	425	450	500	525	600	650	750	900	1050	1350	1800	2700

*For storm sash use one-half of these values.

Allowance for exposure: Add 15% to wall and glass transmission losses also to infiltration losses for wall or walls facing or adjacent to direction of prevailing winds.

If the U values selected are not listed above, the table should be used as follows: Ex: $U^- = .60$. Use column $U = .30$ and multiply results by 2.

To correct for temperatures other than 0°F.-70°F., use Table 35.

To correct for water temperature of 180°F., 190°F. or 200°F., use Tables 36, 37, and 38.

Air Change:

18 square feet $\times$ 1.15 = 20.70 square feet

Roof:

No exposure allowed.

Ceiling	= 11 square feet
Glass skylight	= 7 square feet
Total	= 50.2 square feet

The 50.2 square feet of radiation would maintain the room temperature at 70°F. for 0°F. temperature outside. Inasmuch as the actual conditions are—20°F. to 80°F., Table 34 indicates that the radiation requirements found should be multiplied by 1.57. Therefore, radiation to be installed will be 50.2×1.57 = 79 square feet. Rating tables such as Tables 39 to 45 inclusive are used for this purpose.

Note: Table 12 shows recommended inside temperatures. Table 301 of Appendix shows recommended air changes per hour.

Hot Water Requirements. The standard procedure for hot water systems is carried forward as in Example 1, except that Table 33 is used in place of Table 32 and Tables 35 to 38 inclusive are used (as required) in place of Table 34. The note under each table explains the procedure.

Selections of Radiators. *Tables.** Tables 39 to 47 inclusive are typical radiator rating tables. For example, Table 39 gives dimensions and ratings for

Table 34. Steam. Correction Factors for Temperatures Other Than 0°F.-70°F.

(240 B.t.u. per Square Foot)

Outdoor Temper- ature	Indoor Temperature						
	50	55	60	65	70	75	80
30	.24	.32	.39	.48	.57	.67	.79
25	.30	.38	.46	.55	.64	.75	.87
20	.36	.44	.53	.62	.71	.82	.94
15	.42	.51	.60	.69	.79	.90	1.02
10	.48	.57	.66	.75	.86	.97	1.10
5	.54	.63	.73	.82	.93	1.05	1.18
0	.61	.69	.79	.89	1.00	1.12	1.25
− 5	.67	.76	.86	.96	1.07	1.20	1.33
−10	.73	.82	.92	1.03	1.14	1.27	1.41
−15	.79	.88	.99	1.10	1.21	1.35	1.49
−20	.85	.94	1.05	1.17	1.29	1.42	1.57
−25	.91	1.01	1.12	1.24	1.36	1.50	1.65
−30	.97	1.07	1.18	1.30	1.43	1.58	1.73

Note: Multiply square feet standing radiation required for 0°F.-70°F. by factor from above table to obtain square feet radiation to be installed. See Example 1 of this section.

*Complete tables can be secured from manufacturers.

Table 35. Hot Water. Correction Factors for Temperatures
Other Than 0°F.–70°F.

170° F. Average Water Temperature
(150 B.t.u. per Square Foot)

Outdoor Temper- ature	Indoor Temperature						
	50	55	60	65	70	75	80
30	.23	.30	.38	.47	.57	.68	.81
25	.29	.36	.45	.54	.64	.76	.89
20	.34	.42	.51	.61	.71	.84	.97
15	.40	.48	.58	.68	.79	.92	1.05
10	.46	.54	.64	.74	.86	.99	1.13
5	.52	.60	.70	.81	.93	1.07	1.21
0	.57	.66	.76	.87	1.00	1.14	1.29
− 5	.63	.72	.83	.93	1.07	1.22	1.38
−10	.69	.78	.89	1.01	1.14	1.29	1.46
−15	.75	.84	.96	1.07	1.22	1.37	1.54
−20	.80	.90	1.02	1.14	1.29	1.44	1.62
−25	.86	.96	1.08	1.21	1.36	1.52	1.70
−30	.92	1.02	1.14	1.28	1.43	1.60	1.78

Note: Multiply square feet of standing radiation required for 0°F.–70°F. by factor from above table to obtain square feet radiation to be installed.

Table 36. Hot Water. Correction Factors for Other Than 0°F.–70°F.

180°F. Average Water Temperature

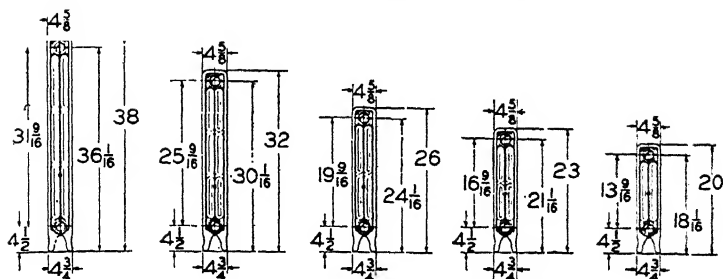| Outdoor Temper- ature	Indoor Temperature						
	50	55	60	65	70	75	80
30	.20	.27	.34	.42	.51	.60	.72
25	.25	.32	.40	.48	.57	.68	.79
20	.30	.37	.45	.54	.63	.75	.87
15	.36	.43	.52	.60	.70	.82	.93
10	.41	.48	.57	.66	.77	.88	1.01
5	.46	.53	.62	.72	.83	.95	1.08
0	.51	.59	.68	.77	.89	1.01	1.15
− 5	.56	.64	.74	.83	.95	1.09	1.23
−10	.61	.69	.79	.90	1.01	1.15	1.30
−15	.67	.75	.85	.95	1.09	1.22	1.37
−20	.71	.80	.91	1.01	1.15	1.28	1.44
−25	.77	.85	.96	1.08	1.21	1.35	1.51
−30	.82	.91	1.01	1.14	1.27	1.42	1.58

Note: If 40 square feet of radiation are required at standard conditions of 0°F.–70°F. and 170°F. Average Water Temperature, and specifications require 0°F.–70°F. and 180°F. Average Water Temperature, multiply 40 by .89 found in above table.

40 × .89 = 35.6 square feet should be installed

If other than 0°F.–70°F. conditions are specified, use above table in same manner.

Table 37. Hot Water. Correction Factors for Other Than 0°F.–70°F.

190°F. Average Water Temperature

Outdoor Temperature	Indoor Temperature						
	50	55	60	65	70	75	80
30	.18	.24	.30	.38	.46	.54	.65
25	.23	.29	.36	.43	.51	.61	.71
20	.27	.34	.41	.49	.57	.67	.78
15	.32	.38	.46	.54	.63	.74	.84
10	.37	.43	.51	.59	.69	.79	.90
5	.42	.48	.56	.65	.74	.86	.97
0	.46	.53	.61	.69	.80	.91	1.03
− 5	.50	.58	.66	.74	.86	.98	1.10
−10	.55	.62	.71	.81	.91	1.03	1.17
−15	.60	.67	.77	.86	.98	1.10	1.23
−20	.64	.72	.82	.91	1.03	1.15	1.30
−25	.68	.77	.86	.97	1.09	1.22	1.36
−30	.74	.82	.91	1.02	1.14	1.28	1.42

Note: If 40 square feet of radiation are required at standard conditions of 0°F.–70°F. and 170°F. Average Water Temperature, and specifications require 0°F.–70°F. and 190°F. Average Water Temperature, multiply 40 × .80 found in above table.

40 × .80 = 32.0 square feet should be installed.

If other than 0°F.–70°F. conditions are specified, use above table in same manner.

Table 38. Hot Water. Correction Factors for Other Than 0°F.–70°F.

200°F. Average Water Temperature

Outdoor Temperature	Indoor Temperature						
	50	55	60	65	70	75	80
30	.17	.22	.27	.34	.41	.49	.58
25	.21	.26	.32	.39	.46	.55	.64
20	.24	.30	.37	.44	.51	.60	.70
15	.29	.35	.42	.49	.57	.65	.76
10	.33	.39	.46	.53	.62	.71	.81
5	.37	.43	.50	.58	.67	.77	.87
0	.41	.47	.55	.63	.72	.82	.93
− 5	.45	.52	.60	.67	.77	.88	.99
−10	.50	.56	.64	.73	.82	.93	1.05
−15	.54	.60	.69	.77	.88	.99	1.11
−20	.58	.65	.73	.82	.93	1.04	1.17
−25	.62	.69	.78	.87	.98	1.09	1.22
−30	.66	.73	.82	.92	1.03	1.15	1.28

Note: If 40 square feet of radiation are required at standard conditions of 0°F.–70°F. and 170°F. Average Water Temperature, and specifications require 0°F.–70°F. and 200°F. Average Water Temperature, multiply 40 × .72 found in above table.

40 × .72 = 28.8 square feet should be installed.

If other than 0°F.–70°F. conditions are specified, use above table in same manner.

Table 39. American Corto Radiators
Three-Tube—Dimensions and Ratings

Width, 4⅝ Inches; Centers, 2½ Inches

Number of Sections	Length 2½ Inches Per Section*	HEATING SURFACE—SQUARE FEET Based upon Engineering Standard of 240 B.t.u. Emission per Square Foot per Hour				
		20-inch Height 1¾ Sq. Ft. Per Section	23-inch Height 2 Sq. Ft. Per Section	26-inch Height 2⅓ Sq. Ft. Per Section	32-inch Height 3 Sq. Ft. Per Section	38-inch Height 3½ Sq. Ft. Per Section
2	5	3½	4	4⅔	6	7
3	7½	5¼	6	7	9	10½
4	10	7	8	9⅓	12	14
5	12½	8¾	10	11⅔	15	17½
6	15	10½	12	14	18	21
7	17½	12¼	14	16⅓	21	24½
8	20	14	16	18⅔	24	28
9	22½	15¾	18	21	27	31½
10	25	17½	20	23⅓	30	35
11	27½	19¼	22	25⅔	33	38½
12	30	21	24	28	36	42
13	32½	22¾	26	30⅓	39	45½
14	35	24½	28	32⅔	42	49
15	37½	26¼	30	35	45	52½
16	40	28	32	37⅓	48	56
17	42½	29¾	34	39⅔	51	59½
18	45	31½	36	42	54	63
19	47½	33¼	38	44⅓	57	66½
20	50	35	40	46⅔	60	70
21	52½	36¾	42	49	63	73½
22	55	38½	44	51⅓	66	77
23	57½	40¼	46	53⅔	69	80½
24	60	42	48	56	72	84
25	62½	43¾	50	58⅓	75	87½
26	65	45½	52	60⅔	78	91
27	67½	47¼	54	63	81	94½
28	70	49	56	65⅓	84	98
29	72½	50¾	58	67⅔	87	101½
30	75	52½	60	70	90	105

TAPPINGS—1½ inches top and bottom. Bushed for steam or water as per specifications. Can be supplied on special order with 6-inch legs, or without legs at no extra charge. For height of loop section only, subtract (3″) inches from total height as shown above.

CONNECTIONS—Both steam and water—extra heavy 1½-inch right and left threaded nipples at top and bottom.

*Add ½ inch to length for each bushing.

Table 40. American Corto Radiators
Four-Tube—Dimensions and Ratings
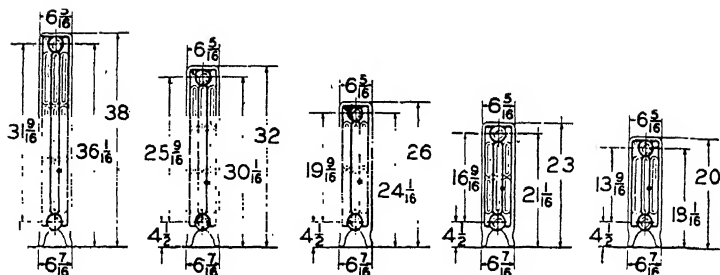
Width, 6⁵⁄₁₆ Inches; Centers, 2½ Inches

Number of Sections	* Length 2½ Inches Per Section	HEATING SURFACE—SQUARE FEET Based upon Engineering Standard of 240 B.t.u. Emission per Square Foot per Hour				
		20-inch Height 2¼ Sq. Ft. Per Section	23-inch Height 2½ Sq. Ft. Per Section	26-inch Height 2¾ Sq. Ft. Per Section	32-inch Height 3½ Sq. Ft. Per Section	38-inch Height 4¼ Sq. Ft. Per Section
2	5	4½	5	5½	7	8½
3	7½	6¾	7½	8¼	10½	12¾
4	10	9	10	11	14	17
5	12½	11¼	12½	13¾	17½	21¼
6	15	13½	15	16½	21	25½
7	17½	15¾	17½	19¼	24½	29¾
8	20	18	20	22	28	34
9	22½	20¼	22½	24¾	31½	38¼
10	25	22½	25	27½	35	42½
11	27½	24¾	27½	30¼	38½	46¾
12	30	27	30	33	42	51
13	32½	29¼	32½	35¾	45½	55¼
14	35	31½	35	38½	49	59½
15	37½	33¾	37½	41¼	52½	63¾
16	40	36	40	44	56	68
17	42½	38¼	42½	46¾	59½	72¼
18	45	40½	45	49½	63	76½
19	47½	42¾	47½	52¼	66½	80¾
20	50	45	50	55	70	85
21	52½	47¼	52½	57¾	73½	89¼
22	55	49½	55	60½	77	93½
23	57½	51¾	57½	63¼	80½	97¾
24	60	54	60	66	84	102
25	62½	56¼	62½	68¾	87½	106¼
26	65	58½	65	71½	91	110½
27	67½	60¾	67½	74¼	94½	114¾
28	70	63	70	77	98	119
29	72½	65¼	72½	79¾	101½	123¼
30	75	67½	75	82½	105	127½

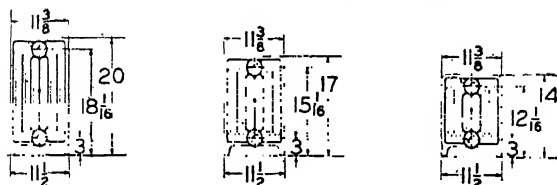TAPPINGS—1½ inches top and bottom. Bushed for steam or water as per specifications.
Can be supplied on special order with 6-inch legs, or without legs at no extra charge.
For height of loop section only, subtract (3″) inches from total height as shown above.
CONNECTIONS—Both steam and water—extra heavy 1½-inch right and left threaded nipples at top and bottom.
*Add ½ inch to length for each bushing.

Table 41. American Corto Radiators

Five-Tube—Dimensions and Ratings
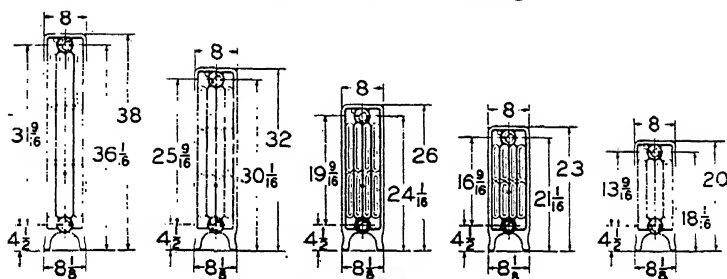
Width, 8 Inches; Centers, 2½ Inches

Number of Sections	*Length 2½ Inches Per Section	HEATING SURFACE—SQUARE FEET Based upon Engineering Standard of 240 B.t.u. Emission per Square Foot per Hour				
		20-inch Height 2⅔ Sq. Ft. Per Section	23-inch Height 3 Sq. Ft. Per Section	26-inch Height 3½ Sq. Ft. Per Section	32-inch Height 4⅓ Sq. Ft. Per Section	38-inch Height 5 Sq. Ft. Per Section
2	5	5⅓	6	7	8⅔	10
3	7½	8	9	10½	13	15
4	10	10⅔	12	14	17⅓	20
5	12½	13⅓	15	17½	21⅔	25
6	15	16	18	21	26	30
7	17½	18⅔	21	24½	30⅓	35
8	20	21⅓	24	28	34⅔	40
9	22½	24	27	31½	39	45
10	25	26⅔	30	35	43⅓	50
11	27½	29⅓	33	38½	47⅔	55
12	30	32	36	42	52	60
13	32½	34⅔	39	45⅓	56⅓	65
14	35	37⅓	42	49	60⅔	70
15	37½	40	45	52½	65	75
16	40	42⅔	48	56	69⅓	80
17	42½	45⅓	51	59½	73⅔	85
18	45	48	54	63	78	90
19	47½	50⅔	57	66½	82⅓	95
20	50	53⅓	60	70	86⅔	100
21	52½	56	63	73½	91	105
22	55	58⅔	66	77	95⅓	110
23	57½	61⅓	69	80½	99⅔	115
24	60	64	72	84	104	120
25	62½	66⅔	75	87½	108⅓	125
26	65	69⅓	78	91	112⅔	130
27	67½	72	81	94½	117	135
28	70	74⅔	84	98	121⅓	140
29	72½	77⅓	87	101½	125⅔	145
30	75	80	90	105	130	150

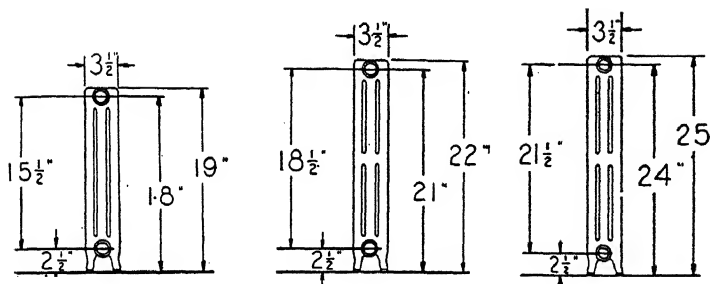TAPPINGS—1½ inches top and bottom. Bushed for steam or water as per specifications.
Can be supplied on special order with 6-inch legs, or without legs at no extra charge.
For height of loop section only, subtract (3″) inches from total height as shown above.
CONNECTIONS—Both steam and water—extra heavy 1½-inch right and left threaded nipples at top and bottom.
*Add ½ inch to length for each bushing.

Table 42. American Corto Window Radiators

Seven-Tube—Dimensions and Ratings

Width, 11⅜ Inches; Centers, 2½ Inches

Number of Sections	* Length 2½ Inches Per Section	Heating Surface—Square Feet Based upon Engineering Standard of 240 B.t.u. Emission per Square Foot per Hour		
		14-inch Height 2½ Sq. Ft. Per Section	17-inch Height 3 Sq. Ft. Per Section	20-inch Height 3⅔ Sq. Ft. Per Section
2	5	5	6	7⅓
3	7½	7½	9	11
4	10	10	12	14⅔
5	12½	12½	15	18⅓
6	15	15	18	22
7	17½	17½	21	25⅔
8	20	20	24	29⅓
9	22½	22½	27	33
10	25	25	30	36⅔
11	27½	27½	33	40⅓
12	30	30	36	44
13	32½	32½	39	47⅔
14	35	35	42	51⅓
15	37½	37½	45	55
16	40	40	48	58⅔
17	42½	42½	51	62⅓
18	45	45	54	66
19	47½	47½	57	69⅔
20	50	50	60	73⅓
21	52½	52½	63	77
22	55	55	66	80⅔
23	57½	57½	69	84⅓
24	60	60	72	88
25	62½	62½	75	91⅔
26	65	65	78	95⅓
27	67½	67½	81	99
28	70	70	84	102⅔
29	72½	72½	87	106⅓
30	75	75	90	110

TAPPINGS—1½ inches top and bottom. Bushed for steam or water as per specifications.
Can be supplied on special order with 4½-inch legs, or without legs at no extra charge.
For height of loop section only, subtract (1½") inches from total height as shown above.
   CONNECTIONS—Both steam and water—extra heavy 1½-inch right and left threaded nipples at top and bottom.
   *Add ½ inch to length for each bushing.

Table 43. Arco Radiators
Three-Tube

Width, 3½ Inches; Centers 1½ Inches

Number of Sections	Length 1½ Inches Per Section	HEATING SURFACE—SQUARE FEET Based upon Engineering Standard of 240 B.t.u. Emission per Square Foot per Hour		
		19-inch Height 1.1 Sq. Ft. Per Section — 0.733 Sq. Ft. Per Lineal Inch	22-inch Height 1.3 Sq. Ft. Per Section — 0.867 Sq. Ft. Per Lineal Inch	25-inch Height 1.5 Sq. Ft. Per Section — 1.000 Sq. Ft. Per Lineal Inch
2	3	2.2	2.6	3
3	4½	3.3	3.9	4.5
4	6	4.4	5.2	6
5	7½	5.5	6.5	7.5
6	9	6.6	7.8	9
7	10½	7.7	9.1	10.5
8	12	8.8	10.4	12
9	13½	9.9	11.7	13.5
10	15	11.0	13.0	15
11	16½	12.1	14.3	16.5
12	18	13.2	15.6	18
13	19½	14.3	16.9	19.5
14	21	15.4	18.2	21
15	22½	16.5	19.5	22.5
16	24	17.6	20.8	24
17	25½	18.7	22.1	25.5
18	27	19.8	23.4	27
19	28½	20.9	24.7	28.5
20	30	22.0	26.0	30
21	31½	23.1	27.3	31.5
22	33	24.2	28.6	33
23	34½	25.3	29.9	34.5
24	36	26.4	31.2	36
25	37½	27.5	32.5	37.5
26	39	28.6	33.8	39
27	40½	29.7	35.1	40.5
28	42	30.8	36.4	42
29	43½	31.9	37.7	43.5
30	45	33.0	39.0	45

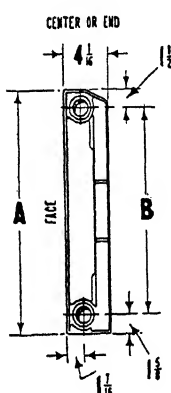CONNECTIONS—Both steam and water extra heavy 1-inch malleable nipples at top and bottom. Can be supplied on special order with 4½-inch legs, or without legs at no extra charge. For height of loop section only, subtract (1½″) inches from total height as shown above. End sections regularly supplied with 1-inch tappings top and bottom, bushed if so specified. Can be tapped 1¼ inches if desired.

Table 44. Arco Radiators
Four-Tube

Width 4¾ Inches; Centers 1½ Inches

Number of Sections	Length 1½ Inches Per Section	HEATING SURFACE—SQUARE FEET Based upon Engineering Standard of 240 B.t.u. Emission per Square Foot per Hour		
		19-inch Height 1.4 Sq. Ft. Per Section	22-inch Height 1.6 Sq. Ft. Per Section	25-inch Height 1.8 Sq. Ft. Per Section
		0.933 Sq. Ft. Per Lineal Inch	1.067 Sq. Ft. Per Lineal Inch	1.200 Sq. Ft. Per Lineal Inch
2	3	2.8	3.2	3.6
3	4½	4.2	4.8	5.4
4	6	5.6	6.4	7.2
5	7½	7.0	8.0	9.0
6	9	8.4	9.6	10.8
7	10½	9.8	11.2	12.6
8	12	11.2	12.8	14.4
9	13½	12.6	14.4	16.2
10	15	14.0	16.0	18.0
11	16½	15.4	17.6	19.8
12	18	16.8	19.2	21.6
13	19½	18.2	20.8	23.4
14	21	19.6	22.4	25.2
15	22½	21.0	24.0	27.0
16	24	22.4	25.6	28.8
17	25½	23.8	27.2	30.6
18	27	25.2	28.8	32.4
19	28½	26.6	30.4	34.2
20	30	28.0	32.0	36.0
21	31½	29.4	33.6	37.8
22	33	30.8	35.2	39.6
23	34½	32.2	36.8	41.4
24	36	33.6	38.4	43.2
25	37½	35.0	40.0	45.0
26	39	36.4	41.6	46.8
27	40½	37.8	43.2	48.6
28	42	39.2	44.8	50.4
29	43½	40.6	46.4	52.2
30	45	42.0	48.0	54.0

CONNECTIONS—Both steam and water extra heavy 1-inch malleable nipples at top and bottom. Can be supplied on special order with 4½-inch legs, or without legs at no extra charge.
For height of loop section only, subtract (1½″) inches from total height as shown above.
End sections regularly supplied with 1-inch tappings top and bottom, bushed if so specified. Can be tapped 1¼ inches if desired.

Table 45. American Fantom Radiators

17-Inch, 20-Inch and 23-Inch One Tube
Width 4¹⁄₁₆ Inches; Centers 2½ Inches

DIMENSIONS—ONE TUBE

	A	B
17″	17¹⁄₁₆	13¹⁵⁄₁₆
20″	20	16¹⁵⁄₁₆
23″	22¹³⁄₁₆	19¹¹⁄₁₆

TAPPINGS—1¼ inches top and bottom. Bushed for steam or water as per specifications.

CONNECTIONS—Both steam and water—extra heavy 1¼-inch right and left threaded nipples at top and bottom.

Number of Sections	* Length 2½ Inches Per Section	† 17 Inches Height 1¾ Square Feet Per Section	† 20 Inches Height 2 Square Feet Per Section	† 23 Inches Height 2¼ Square Feet Per Section
2	5	3½	4	4½
3	7½	5¼	6	6¾
4	10	7	8	9
5	12½	8¾	10	11¼
6	15	10½	12	13½
7	17½	12¼	14	15¾
8	20	14	16	18
9	22½	15¾	18	20¼
10	25	17½	20	22½
11	27½	19¼	22	24¾
12	30	21	24	27
13	32½	22¾	26	29¼
14	35	24½	28	31½
15	37½	26¼	30	33¾
16	40	28	32	36
17	42½	29¾	34	38¼
18	45	31½	36	40½
19	47½	33¼	38	42¾
20	50	35	40	45
21	52½	36¾	42	47¼
22	55	38½	44	49½
23	57½	40¼	46	51¾
24	60	42	48	54
25	62½	43¾	50	56¼

*Add ½ inch to length for each bushing.
†Based on engineering standards of 215 degrees Fahrenheit steam temperature, 70 degrees room temperature and 240 B.t.u. per square foot per hour.
　　Regularly furnished without legs, but can be supplied on special order with 4½″ legs only at no extra charge.

Table 46. Arco Convector Steam or Water Ratings

Steam or Water—No. 3—3¾" Wide
*Output in Sq. Ft. Equivalent Direct Radiation

"L" Length Inches	13	18	20½	23	25½	28	30½	33	35½	38	40½	43	45½	53	55½	58	60½
Unit No.	313	318	320	323	325	328	330	333	335	338	340	343	345	353	355	358	360
ENCLOSURE HEIGHT 18	6.0	8.4	9.7	10.9	12.1	13.3	14.5	15.8	17.0	18.2	19.4	20.6	21.8	25.5	26.7	28.0	29.2
20	6.5	9.1	10.5	11.8	13.1	14.4	15.7	17.1	18.4	19.7	21.0	22.3	23.6	27.6	28.9	30.3	31.6
22	7.0	9.8	11.2	12.7	14.1	15.5	16.9	18.4	19.8	21.3	22.6	24.1	25.4	29.8	31.1	32.6	34.0
24	7.5	10.5	12.0	13.5	15.0	16.5	18.1	19.7	21.1	22.7	24.2	25.7	27.2	31.8	33.3	34.8	36.3
26	7.8	11.0	12.6	14.2	15.8	17.4	19.0	20.6	22.2	23.8	25.3	26.9	28.5	33.3	34.9	36.5	38.1
29	8.2	11.5	13.1	14.8	16.4	18.1	19.8	21.5	23.1	24.8	26.4	28.1	29.7	34.7	36.3	38.0	39.6
32	8.4	11.8	13.5	15.2	16.9	18.6	20.4	22.1	23.8	25.5	27.2	28.9	30.6	35.7	37.4	39.1	40.8
35	8.6	12.1	13.8	15.6	17.3	19.0	20.8	22.6	24.3	26.1	27.8	29.6	31.3	36.5	38.2	40.0	41.7
38	8.8	12.3	14.1	15.9	17.6	19.4	21.2	23.0	24.7	26.5	28.3	30.1	31.8	37.2	38.9	40.7	42.5
42	8.9	12.5	14.3	16.1	17.9	19.8	21.6	23.4	25.2	27.0	28.8	30.6	32.4	37.9	39.7	41.5	43.3
47	9.1	12.8	14.6	16.5	18.3	20.2	22.0	23.9	25.7	27.6	29.4	31.3	33.1	38.7	40.5	42.4	44.2
57	9.4	13.2	15.1	17.0	18.9	20.8	22.7	24.6	26.5	28.4	30.3	32.2	34.1	39.8	41.7	43.6	45.5

"L" Length Inches	63	65½	68	70½	73	75½	78	80½	83	85½	88	95½	98	100½
Unit No.	363	365	368	370	373	375	378	380	383	385	388	395	398	3100
ENCLOSURE HEIGHT 18	30.5	31.7	33.0	34.2	35.5	36.7	38.0	39.2	40.5	41.7	43.0	46.7	48.0	49.2
20	33.0	34.3	35.7	37.0	38.4	39.7	41.1	42.4	43.8	45.1	46.5	50.5	51.9	53.2
22	35.5	36.9	38.4	39.8	41.3	42.7	44.2	45.6	47.1	48.5	50.0	54.3	55.8	57.2
24	37.9	39.4	41.0	42.5	44.1	45.6	47.2	48.7	50.3	51.8	53.4	58.0	59.6	61.1
26	39.7	41.3	42.9	44.5	46.1	47.7	49.3	50.9	52.5	54.1	55.7	60.5	62.1	63.7
29	41.3	42.9	44.6	46.2	47.9	49.5	51.2	52.8	54.5	56.1	57.8	62.7	64.4	66.0
32	42.5	44.2	45.9	47.6	49.3	51.0	52.7	54.4	56.1	57.8	59.5	64.6	66.3	68.0
35	43.5	45.2	47.0	48.7	50.5	52.2	54.0	55.7	57.5	59.2	61.0	66.2	68.0	69.7
38	44.3	46.1	47.9	49.7	51.5	53.3	55.1	56.9	58.7	60.5	62.3	67.7	69.5	71.3
42	45.1	46.9	48.7	50.5	52.3	54.1	55.9	57.7	59.5	61.3	63.1	68.5	70.3	72.1
47	46.1	47.9	49.8	51.6	53.5	55.3	57.2	59.0	60.9	62.7	64.6	70.1	72.0	73.8
57	47.4	49.3	51.2	53.1	55.0	56.9	58.8	60.7	62.6	64.5	66.4	72.1	74.0	75.9

Convector lengths are shown. Standard enclosures are 1½" longer.
Above ratings based on front outlet enclosures. For top outlet ratings see opposite page.
Enclosure Heights Are Stack Height Plus 7"
*To determine size of radiator divide total heat loss in B.t.u. by 240 for steam at 215°, or by 150 B.t.u. for water at 170°.

Table 47. Arco Convector Steam or Water Ratings

Steam or Water—No. 5—5⅝″ Wide
*Output in Sq. Ft. Equivalent Direct Radiation

"L" Length Inches	13	18	20½	23	25½	28	30½	33	35½	38	40½	43	45½	53	55½	58	60½
Unit No.	513	518	520	523	525	528	530	533	535	538	540	543	545	553	555	558	560
ENCLOSURE HEIGHT 18	8.7	12.3	14.1	15.9	17.7	19.5	21.3	23.1	24.9	26.7	28.5	30.3	32.1	37.5	39.3	41.1	42.9
20	9.5	13.5	15.4	17.4	19.3	21.3	23.2	25.2	27.2	29.2	31.1	33.1	35.0	40.9	42.8	44.8	46.7
22	10.3	14.5	16.6	18.7	20.8	23.0	25.1	27.2	29.3	31.4	33.5	35.6	37.7	44.1	46.2	48.3	50.4
24	10.9	15.4	17.6	19.9	22.2	24.4	26.6	28.9	31.1	33.4	35.6	37.9	40.1	46.8	49.0	51.3	53.5
26	11.5	16.2	18.5	20.9	23.2	25.6	28.0	30.4	32.7	35.1	37.4	39.8	42.1	49.2	51.5	53.9	56.3
29	12.0	17.0	19.4	21.9	24.4	26.9	29.3	31.8	34.3	36.8	39.2	41.7	44.2	51.6	54.0	56.5	59.0
32	12.4	17.5	20.1	22.6	25.2	27.7	30.3	32.9	35.4	38.0	40.5	43.1	45.6	53.3	55.8	58.4	60.9
35	12.7	17.9	20.5	23.1	25.7	28.3	30.9	33.5	36.1	38.7	41.3	43.9	46.5	54.3	56.9	59.5	62.1
38	12.9	18.2	20.8	23.5	26.1	28.8	31.4	34.1	36.7	39.4	42.0	44.7	47.3	55.3	57.9	60.6	63.2
42	13.2	18.6	21.3	24.0	26.7	29.4	32.1	34.8	37.5	40.2	42.9	45.6	48.4	56.5	59.2	61.9	64.6
47	13.5	19.0	21.8	24.5	27.2	30.0	32.8	35.6	38.3	41.1	43.8	46.6	49.4	57.7	60.4	63.2	65.9
57	13.8	19.5	22.4	25.2	28.1	30.9	33.8	36.6	39.4	42.3	45.1	48.0	50.8	59.4	62.2	65.1	67.9

"L" Length Inches	63	65½	68	70½	73	75½	78	80½	83	85½	88	95½	98	100½
Unit No.	563	565	568	570	573	575	578	580	583	585	588	595	598	5100
ENCLOSURE HEIGHT 18	44.7	46.5	48.3	50.1	51.9	53.7	55.5	57.3	59.1	60.9	62.7	68.1	69.9	71.7
20	48.7	50.6	52.6	54.5	56.5	58.4	60.4	62.3	64.3	66.2	68.2	74.0	76.0	77.9
22	52.5	54.6	56.7	58.8	60.9	63.0	65.1	67.2	69.3	71.4	73.5	79.8	81.9	84.0
24	55.8	58.0	60.3	62.5	64.8	67.0	69.3	71.5	73.8	76.0	78.3	85.0	87.3	89.5
26	58.7	61.0	63.4	65.7	68.1	70.4	72.8	75.1	77.5	79.8	82.2	89.2	91.6	93.9
29	61.5	63.9	66.4	68.8	71.3	73.7	76.2	78.6	81.1	83.5	86.0	93.3	95.8	98.2
32	63.5	66.0	68.6	71.1	73.7	76.2	78.8	81.3	83.9	86.4	89.0	96.6	99.2	101.7
35	64.7	67.3	69.9	72.5	75.1	77.7	80.3	82.9	85.5	88.1	90.7	98.5	101.1	103.7
38	65.9	68.5	71.2	73.8	76.5	79.1	81.8	84.4	87.1	89.7	92.4	100.3	103.0	105.6
42	67.3	70.0	72.7	75.4	78.1	80.8	83.5	86.2	88.9	91.6	94.3	102.4	105.1	107.8
47	68.7	71.4	74.2	76.9	79.7	82.4	85.2	87.9	90.7	93.4	96.2	104.4	107.2	109.9
57	70.8	73.6	76.5	79.3	82.2	85.0	87.9	90.7	93.6	96.4	99.3	107.8	110.7	113.5

Above ratings based on front outlet enclosures. Top outlet ratings are given below.

Enclosure Heights Are Stack Height Plus 7″

*To determine size of radiator divide total heat loss in B.t.u. by 240 for steam at 215°, or by 150 B.t.u. for water at 170°.

Top Outlet Rating Table

Where top outlet grilles are used in enclosures—output ratings may be increased as follows:

18″ enclosure heights multiply ratings by 1.15
20″ enclosure heights multiply ratings by 1.10
22″ enclosure heights multiply ratings by 1.06
24″ enclosure heights multiply ratings by 1.04
26″ enclosure heights multiply ratings by 1.02

No increase beyond 26″ enclosure heights.

20, 23, 26, 32, and 38-inch three-tube American Corto radiators according to the number of sections. Thus a three-tube American Corto having 10 sections and standing 20 inches high is rated at 17½ square feet of heating surface.

Example 2. Select a three-tube Corto radiator to fulfil requirements of Example 1.

In Example 1 it was determined that 72 square feet of radiation were required. Studying Table 39 we see that the 20, 23, and 26-inch height radiators, even with 30 sections, would not give 72 square feet. Either a 32 or a 38-inch radiator can be used because a 32-inch having 24 sections gives exactly 72 square feet, and a 38-inch having 21 sections gives 73½ square feet. If both the 32 and 38-inch radiators were too high and if floor space were plentiful, two 20-inch radiators having 21 sections, two 23-inch radiators having 18 sections, or two 26-inch radiators having 16 sections could be used, and would give the required heating surface. A 17-inch seven-tube window radiator having 24 sections would fulfil requirements of Example 1 also.

Other types and sizes are selected in the same way. Any one requirement can generally be satisfied by one of several heights, tubes, sections, etc. However, the position of the radiator, its location, how it is used, etc., must be considered.

Effect of Painting. The table below shows the relative heating value of radiators with four different finishes.

†Effect of Painting 32=inch Three Column, Six=Section Cast=Iron Radiator

‡ Radiator No.	Finish	Area Square Feet	Relative Heating Value Per Cent
1	Bare iron, foundry finish	27	100.5
2	One coat of aluminum bronze	27	90.8
3	Grain paint dipped	27	101.1
4	One coat dull black Pecora paint	27	100.0

*Complete tables can be secured from manufacturers.
†A.S.H.V.E. Guide, 1936, Chapter 30.
‡All radiators are column types.

Selection of Pipe Coils. If pipe coils are to be used, it becomes necessary to reduce square feet of heating surface to linear feet of pipe. This can be done by means of the following factors:

Square feet of heating surface $\times$
$\begin{cases} =\text{linear feet of } 1 \text{ -inch pipe} \\ =\text{linear feet of } 1\frac{1}{4}\text{-inch pipe} \\ =\text{linear feet of } 1\frac{1}{2}\text{-inch pipe} \\ 1.6 =\text{linear feet of } 2 \text{ -inch pipe} \end{cases}$

The size of radiator is sufficient to keep the room warm only after it is heated. No allowance is made for warming it. In other words, the heat given off by the radiator is equal only to that lost through walls and windows. This condition is offset in two ways—when (1) the room is cold, the difference in temperature between the steam and the air of the room is greater, and the radiator is more efficient; (2) the radiator is proportioned for the coldest weather, so that for a greater part of the time it is larger than necessary.

PRACTICE PROBLEMS
Use Method (A)

1. Note Figs. 69 and 70. In this example, assume the following structural specifications.

Walls: 2×4 frame. Bildrite sheathing (⅞-inch) is used in place of wood sheathing. Lok-Joint (½-inch) used in place of lath. The outside is shingle covered and there is plaster on the inside.
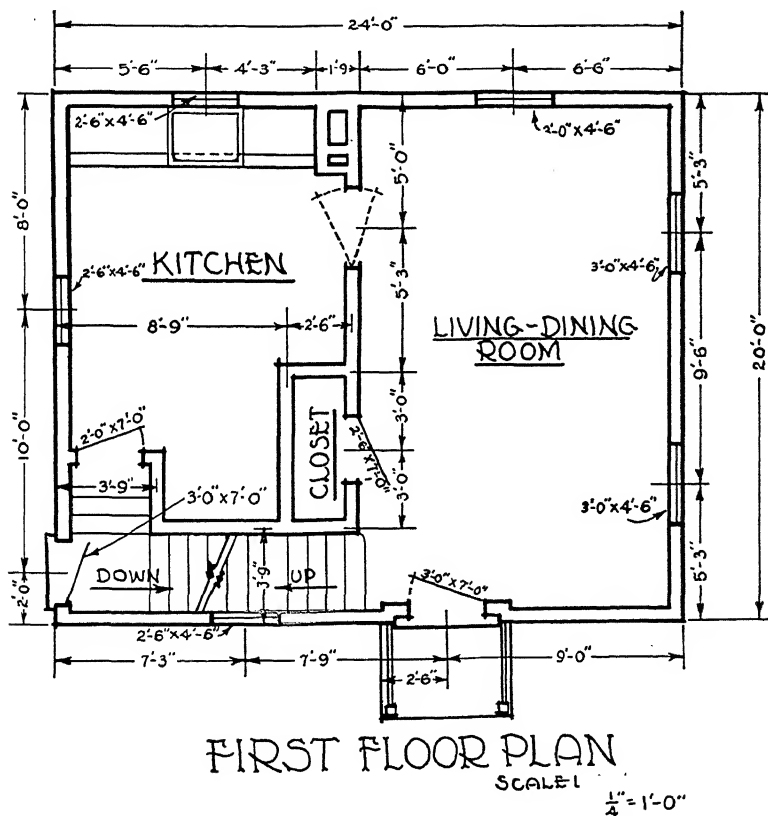
Fig. 69. First Floor Plan

Windows: All single glass with weatherstripping.

Doors: Not weatherstripped.

First Floor: Joists, ½-inch Insulite; ⅝-inch finish flooring.

Second Floor Ceiling: Plaster on 1-inch Lok-Joint lath; joists, ⅝-inch rough pine floor.

Basement: Assume basement is warm.

Attic: Assume attic temperature same as outside.

The house faces due north. Assume inside temperature as 72°F. and outside temperature −23°F. (coldest on record).

How many square feet of steam radiation will be required to maintain a temperature of 72°F. in all first and second floor rooms?

2. Select American Corto radiators of the four-tube type for each room of the house in Problem 1. The height of the various registers for each room is left to the reader's judgment.
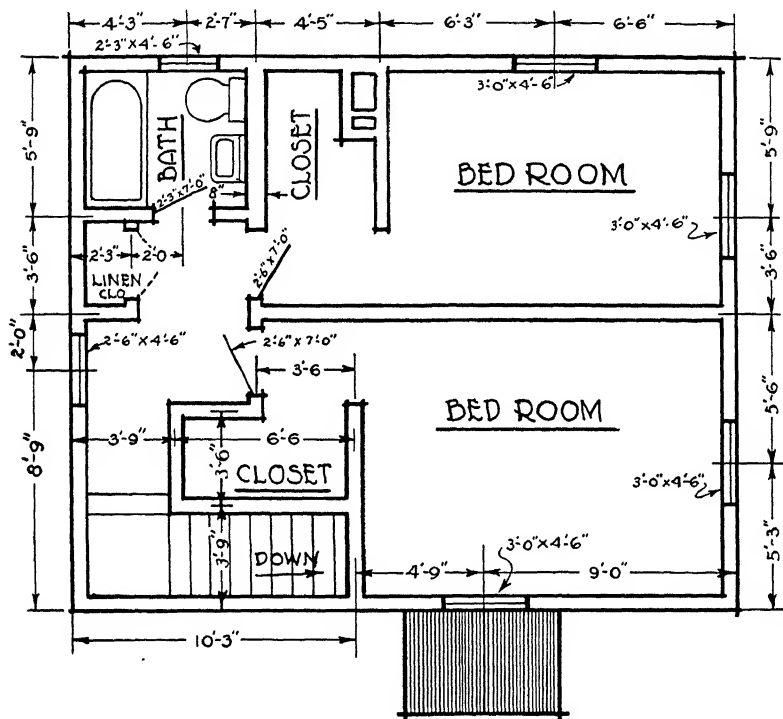
Fig. 70. Second Floor Plan

3. Make a sketch of the house in Problem 1 and, representing radiators by small rectangles, indicate the best position for radiators in all of the rooms. Give reasons for the positions.

Method (B). Radiator Requirements by B.t.u. Method. In some cases, instead of using the foregoing method of calculating radiator requirements directly, it is advantageous to calculate the heat losses in each room of the given structure and from these determine the radiator requirements.

Heat Transmission of Direct=Steam Radiators. The heat transmitted by a radiator, that is, the B.t.u. which it gives off per square foot of surface per hour, depends upon the difference in temperature between the steam in the radiator and the surrounding air, the velocity of air currents above the radiator, and the smooth or rough quality of the radiator surface. In ordinary low-pressure heating, the first condition is practically constant, but the second varies somewhat with the pattern of the radiator. An open design which allows the air to circulate freely over the radiating surfaces is more efficient than a closed pattern, and for this reason a pipe coil is more efficient than a radiator.

Coefficient of Heat Transmission. The coefficient of heat transmission is the heat given off per square foot of radiating surface per hour for each degree difference in temperature between the steam within the radiator and the air surrounding it. Coefficients of heat transmission for various types and heights of black iron radiators are given in Table 48. These coefficients are not applicable to column radiators of three or four sections only.

Coefficients are the result of tests with a steam temperature of 220°F. and an air temperature of 70°F. For steam and air temperature differences less than 150°F., the coefficients are slightly smaller, and for temperature differences greater than 150°F., the coefficients are slightly greater than those given in Table 48.

Table 48. *Coefficient of Heat Transmission (K) for Direct=Steam Radiators

Type of Radiator	Height of radiator (inches)			
	20–22	26	32	38
1 Column...........................	1.95	1.90	1.85	1.80
2 Columns..........................	1.80	1.75	1.70	1.65
3 Columns..........................	1.70	1.65	1.60	1.55
4 Columns..........................	1.60	1.55	1.50	1.45
Window............................	1.85	....	....	....
Pipe coils..........................	2.00	....	....	....
Wall, horizontal.....................	1.95	....	....	....
Wall, vertical.......................	1.90	....	....	....

*Mechanical Equipment of Buildings, Volume I, by Harding and Willard.

From the above it is easy to compute the radiator size for any given room. First, compute the heat loss per hour by conduction and leakage in the coldest weather; then divide the result by the heat transmission per square foot, per hour, for the type of radiator to be used.

Thus with a 3-column 38-inch cast-iron radiator standing in air at 70°F.

and filled with steam at about 3 pounds per square inch pressure, or 220°F., the heat transmission per square foot of radiating surface per hour is calculated as follows:

Steam and air temperature difference = 220−70 = 150°F. Heat transmitted per square foot of radiating surface per hour = 1.55×150 = 232.5 B.t.u. radiation factor. The heat that 100 square feet of the above radiation will supply per hour is 100×232.5 = 23,250 B.t.u.

Table 48, as already stated, is based on the average performance of direct radiators in rooms where the air is at 70°F. and steam at 220°F. The temperature difference is then 150°F. and is called standard. Sometimes conditions not standard are encountered, in which case some adjustment must be made to the K value taken from Table 48. Thus, if a three-column 38-inch direct radiator is used in a room where the temperature is maintained at 60°F., with steam temperature at 230°F., we have a temperature range of 170°F. or 20°F. above the standard condition, and the value of K becomes

$$*K = (1.55 + 0.002 \times 20 \times 1.55) = 1.61$$

Thus each square foot of radiation would give off 1.61×170 = 274 B.t.u. per hour.

Heat Transmission of Direct Hot-Water Radiators. Table 48 can also be used for K values for hot-water radiators. Allowance must be made for the lower temperature range in hot-water heating. Thus, with a room usually at 70°F., and water at 180° entering and 160°F. leaving the radiator, the temperature range is 100°F., or 50°F. less than the standard. Thus, for a two-column 26-inch direct radiator the value of K is

$$K = (1.75 - 0.002 \times 50 \times 1.75) = 1.58$$

and each square foot of this radiation gives off 1.58×100 = 158 B.t.u. per hour.

The usual assumptions made for the heat transmission of direct radiation are 240 B.t.u. per square foot per hour for low pressure steam cast-iron radiators and 150 B.t.u. per square foot per hour for cast-iron hot-water radiators with the water at 180°. Table 48 is used for more exact values. Thus a hot-water system requires 60% more radiation than a low pressure steam system.

Example. Find the amount of direct cast-iron radiation, low pressure steam and hot water, to supply a heat loss of 10,000 B.t.u. per hour.

Steam: 10,000÷240 = 42 square feet.

Hot Water: 10,000÷150 = 66⅔ square feet.

If a three-column radiator 38 inches high is used, the heating surface of which is 5 square feet per section, it will require 42÷5 = 8 sections (approx.) for the steam system, making radiator length equal to 8x2½ = 20 inches.

Electric Radiator Valves. Fig. 71 shows a typical electric radiator valve which is used to control the flow of steam or hot water to individual radiators. It can be operated either singly or in groups, usually under the control of a room thermostat. In addition to zone control, such controls make possible the individual heating of rooms

*Harding & Willard. Vol. 1.

where, for example, rooms on the windward side of a residence need more steam than the warmer side away from the wind.

The valves are operated by a small induction motor driven through a gear train to open or close the valves as the thermostat
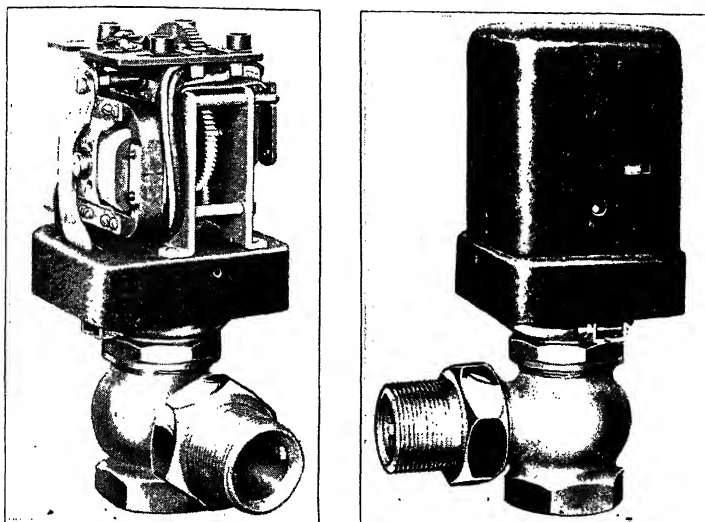
Fig. 71. Interior and Exterior Views of an Electric Radiator Valve
Courtesy of Minneapolis Honeywell Regulator Company

demands. They are provided also with a shaft extension which permits manual operation if required during power interruption, etc.

Motorized Valves. Fig. 72 shows a typical motorized valve used, for instance, under command of a modulating room thermostat or duct controller. The motorized valve may be used to control automatically low pressure steam to the heating coils of a heating, ventilating, or air-conditioning system in varying quantities, as the demand varies in a room or enclosure.

Magnetic Shutoff Valves. Fig. 73 shows a sectional view of a typical magnetic shutoff valve. This is used especially for residence air washers but can be used also in controlling other non-corrosive liquids.

The tabulation shown in Table 49 gives flow characteristics. The table shows, for example, that at 2 pounds water pressure .77 gallon per minute flows through the valve.
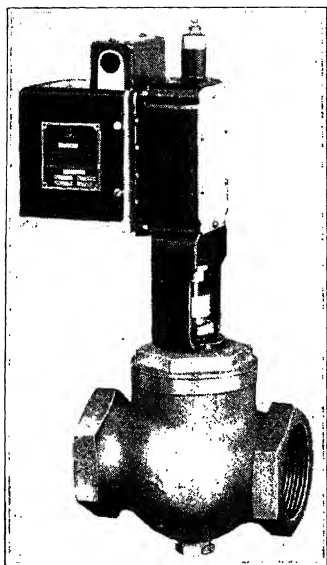
Fig. 72. Motorized Valve
Courtesy of Minneapolis Honeywell Regulator Company
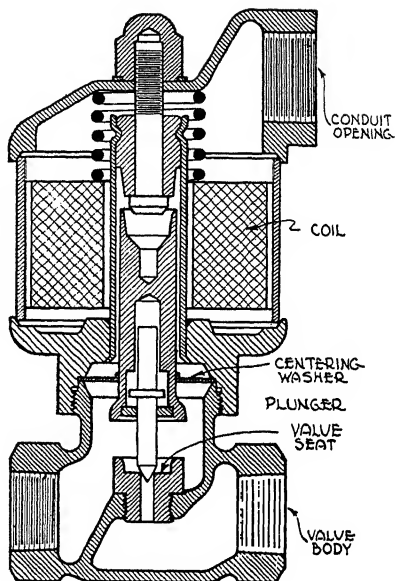
Fig. 73. Section of Magnetic Shutoff Valve

Table 49. Flow Characteristics

Pressure per Sq. In.	Gallons per Minute	Pressure per Sq. In.	Gallons per Minute	Pressure per Sq. In.	Gallons per Minute
2 lbs.	.77	20 lbs.	1.87	75 lbs.	2.83
6 lbs.	1.25	30 lbs.	2.03	100 lbs.	3.29
10 lbs.	1.50	40 lbs.	2.26	125 lbs.	3.62
15 lbs.	1.74	50 lbs.	2.45	150 lbs.	3.75

PRACTICE PROBLEMS

Use Method (B)

1. A room has a heat loss of 32,560 B.t.u. per hour in the coldest weather. The inside temperature is 68°F. and the steam temperature 240°F. What amount will be required of 4-tube (use Table 40) 26-in. high, cast-iron radiator?

2. The heat loss from a room at 70°F. is 25,000 B.t.u. per hour in the coldest weather. Steam at 222°F. What amount will be required of direct 3-tube 32-inch high, cast-iron radiator?

3. A schoolroom is to be warmed with circulation coils of 1¼-inch pipe. The heat loss is 30,000 B.t.u. per hour at 70°F. Steam temperature 220°F. What length of pipe will be required?

Note: No tables for column radiators are shown. However, they can be secured from manufacturers upon special request.

Summary. Two methods of determining radiator requirements have been given in this section. Method (A) is the estimating method by which good results may be obtained if the structural types being considered are within its limitations. While some engineers might disapprove of Method (A), it is really a helpful and rapid way to estimate radiator requirements and has proved practical.

Method (B) is more accurate perhaps especially if all of its unit parts are expressed accurately as regards total heat losses, K values, etc.

The B.t.u. method has the approval of the American Society of Heating and Ventilating Engineers as an accurate procedure.

LAYOUT OF A TYPICAL WEBSTER VACUUM SYSTEM OF STEAM HEATING
USING LOW PRESSURE STEAM BOILER AND MOTOR-DRIVEN VACUUM PUMP
THIS IS ONLY ONE OF MANY POSSIBLE ARRANGEMENTS. THE FLEXIBILITY OF WEBSTER VACUUM SYSTEMS
PERMITS MANY VARIATIONS IN PIPING ARRANGEMENTS, RADIATION AND OTHER EQUIPMENT.

Courtesy of Warren Webster & Co., Camden, N. J.

CHAPTER VIII

PORTABLE RADIATORS

There are many types and makes of portable radiators which can be used in various locations.

Electric Steam Radiator. Fig. 74 shows a portable electric steam radiator mounted on ball-bearing casters. Because of the casters,
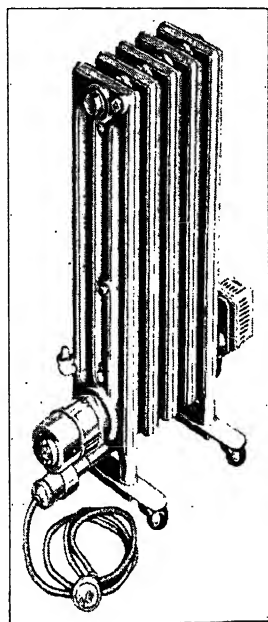
Fig. 74. "Heat-On-Wheels" Portable Electric Steam Radiator
Courtesy of Burnham Boiler Corporation, Irvington, N.Y.

the radiator can be moved from room to room and is ready for instant use. The heating element uses electricity which can be supplied from any 110-volt alternating current outlet. The heating elements for the four sizes manufactured range from 750 to 1,500 watts. The radiator is equipped with a thermostat capable of temperature settings between 55° and 80° F. Anti-freeze solution mixed with water prevents freezing when the radiator is not in use.

The radiator shown in Fig. 74 can be obtained in four sizes, which have heating effects of 10½, 14, 17½, and 21 square feet. The selection may be made on the same basis as that explained for direct-steam radiators.

Gas Steam Radiator. In stores, restaurants, garages, and other
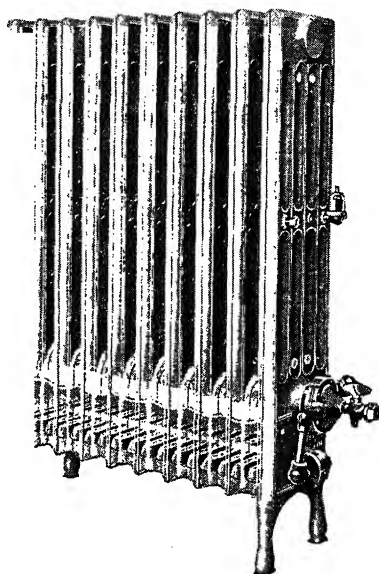
Fig. 75. Pittsburgh Gas-Steam Radiator
Courtesy of Burnham Boiler Corporation, Irvington, N.Y.

places where the installation of a central heating plant would be too expensive, a gas steam radiator can be used to advantage.

Pittsburgh Gas Steam Radiator. The gas steam radiator shown in Fig. 75 operates on common gas the same as that used for cooking. The gas is mixed with air exactly as on a gas stove. The burners are at the bottom of the radiator, as shown in Fig. 75. Water is supplied by hand, or by automatic valves when water pipes are attached to the radiator. The radiator operates and heats like an ordinary direct steam radiator.

The selection of the correct size is done exactly as explained for direct steam radiators. Table 50 shows data and ratings for 5- and 7-tube types.

Table 50. Data and Ratings for Pittsburgh Gas-Steam Radiator

| Number of Sections | | Capacity, Square Feet, When Used With | | Length, Inches | Weight, Lbs. |
		Gas	Regular Heating System		
5-Tube—26 Inches High—Unvented					
EL	4	24	10 $\frac{2}{3}$	15	100
EL	5	29	13 $\frac{1}{3}$	17 $\frac{1}{2}$	120
EL	6	34	16	20	138
EL	8	44	21 $\frac{1}{3}$	25	184
EL	10	54	26 $\frac{2}{3}$	30	230
EL	12	64	32	35	276
EL	15	79	40	42 $\frac{1}{2}$	345
5-Tube—38 Inches High—Unvented					
EH	4	37	17 $\frac{1}{3}$	15	130
EH	5	45	21 $\frac{2}{3}$	17 $\frac{1}{2}$	155
EH	6	52	26	20	180
EH	8	66	34 $\frac{2}{3}$	25	240
EH	10	79	43 $\frac{1}{3}$	30	300
7-Tube—27 Inches High—Unvented					
LW	5	43	18 $\frac{1}{3}$	17 $\frac{1}{2}$	120
LW	6	51	22	20	144
LW	7	58	25 $\frac{2}{3}$	22 $\frac{1}{2}$	168
LW	8	66	29 $\frac{1}{3}$	25	192
LW	9	74	33	27 $\frac{1}{2}$	216
LW	10	82	36 $\frac{2}{3}$	30	240
7-Tube—32 Inches High					
HW	5	55	23 $\frac{3}{4}$	17 $\frac{1}{2}$	175
HW	6	64	28 $\frac{1}{2}$	20	205
HW	7	74	33 $\frac{1}{4}$	22 $\frac{1}{2}$	245
HW	8	83	38	25	275
HW	9	93	42 $\frac{3}{4}$	27 $\frac{1}{2}$	310
HW	10	102	47 $\frac{1}{2}$	30	350

TYPICAL ELEVATION OF WEBSTER TYPE "R" SYSTEM OF STEAM HEATING

Courtesy of Warren Webster & Co., Camden, N. J.

CHAPTER IX

DIRECT=STEAM HEATING

A system of direct-steam heating includes a furnace and boiler* for the generation of steam, radiators†, pipes, and fittings. This section describes the parts of a direct-steam system with the exception of boilers and radiators.

Steam systems may incorporate either a gravity or a mechanical return application. In gravity systems, the condensate is returned to the boiler by gravity due to the static head of water in the return mains. The elevation of the boiler water-line consequently must be sufficiently below the lowest heating units and steam main and dry return mains to permit the return of condensate by gravity. The distance between the water-line of the boiler and the level of the water in the dry or wet return main (water-line difference) must be sufficient to overcome the maximum pressure drop in the system.‡

In mechanical systems, the condensate flows to a receiver and is then forced into the boiler against the boiler pressure. The lowest parts of the supply side of the system must be kept sufficiently above the water-line of the receiver to insure adequate drainage of water from the system. The relative elevation of the boiler water-line, however, is unimportant in such cases except that the head on the pump or trap discharge becomes greater as the height of the boiler water-line above the trap or pump increases.

Systems of Piping. There are three systems of piping: (1) the two-pipe system, (2) the one-pipe relief system, and (3) the one-pipe circuit system, with modifications and combinations of each system.

Two=Pipe System. Fig. 76 shows the arrangement of piping and radiators in the two-pipe system. The steam main leads from the top of the boiler, and the branches are carried along near the basement ceiling. Risers are taken from the supply branches, and carried up to the radiators on the different floors; and return pipes are

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*Data relative to boilers is in Chapter V.
†Data relative to radiators (for steam, vapor or hot water) is in Chapter VII.
‡For a detailed presentation of piping see "Steam and Hot Water Fittings," published by American Technical Society, Chicago.

brought down to the return mains, which should be placed near the basement floor below the water-line of the boiler. Where the building is more than two stories high, radiators in similar positions on different floors are connected with the same riser, which may run to the highest floor; and a corresponding return drop connecting with each radiator is carried down beside the riser to the basement. A system in which the main horizontal returns are below the water-
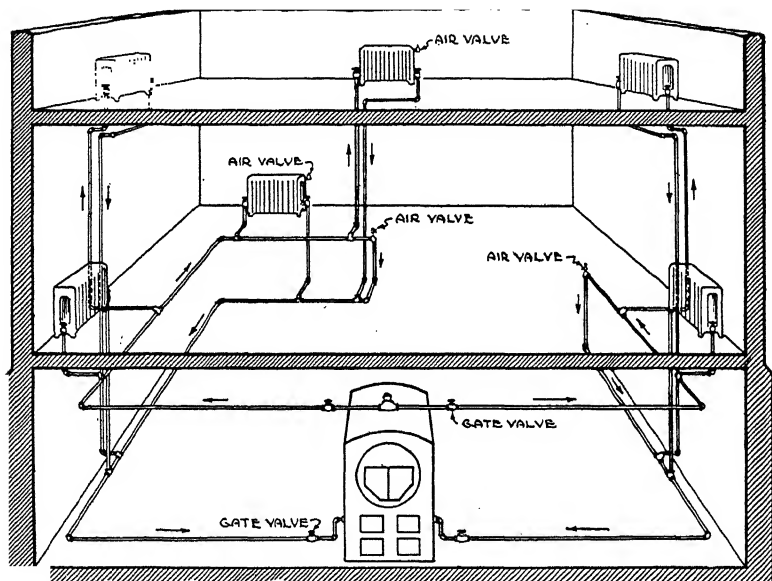
Fig. 76. Arrangement of Piping and Radiators in "Two-Pipe" System (Now Obsolete)

line of the boiler is said to have a wet or sealed return. If the returns are overhead and above the water-line, it is called a dry return. Where the steam is exposed to extended surfaces of water, as in overhead returns, where the condensation partially fills the pipes, there is likely to be cracking or water-hammer, due to the sudden condensation of the steam as it comes in contact with the cooler water. This is especially noticeable when steam is first turned into cold pipes and radiators, and the condensation is excessive. When dry returns are used, the pipes should be large and have a good pitch toward the boiler.

In the case of sealed returns, the only contact between the steam and standing water is in the vertical returns, where the exposed sur-

faces are very small (being equal to the sectional area of the pipes), and trouble from water-hammer is practically eliminated. Dry returns should be given an incline of at least 1 inch in 10 feet, while for wet returns 1 inch in 20 or even 40 feet is ample. The ends of all steam mains and branches should be dripped into the returns. If the return is sealed, the drip may be directly connected as shown in Fig. 77; but if it is dry, the connection should be provided with a siphon loop as indicated in Fig. 78. The loop becomes filled with water and thus prevents steam from flowing directly into the return. As the condensation collects in the loop, it overflows into the return pipe and is carried away. The return pipes in this case are of course
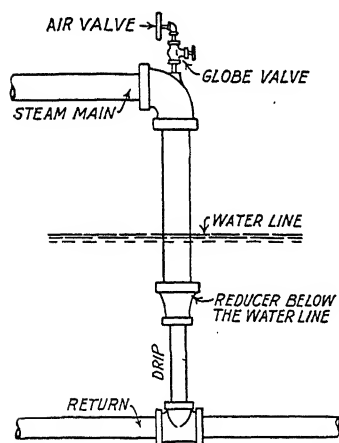
Fig. 77. End of Steam Main Connected to a Wet Return
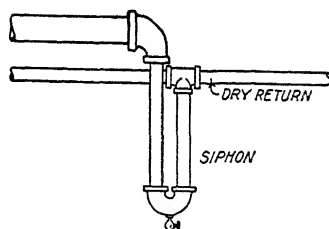
Fig. 78. Use of Siphon in Connecting Drip to a Dry Return

filled with steam above the water; but it is steam which has passed through the radiators and their return connections, and is therefore at a slightly lower pressure; so that, if steam were admitted directly from the main, it would tend to hold back the water in more distant returns and cause surging and cracking in the pipes. Sometimes the boiler is at a lower level than the basement in which the returns are run, and it then becomes necessary to establish a false water-line. This is done by making connections as shown in Fig. 79.

It is readily seen that the return water, in order to reach the boiler, must flow through the trap, which raises the water-line or

seal to the level shown by the dotted line. The balance pipe is to
equalize the pressure above and below the water in the trap, and
prevent siphonic action, which would tend to drain the water out of
the return mains after a flow was once started.

The balance pipe, when possible, should be 15 or 20 feet in
length, with a throttle-valve placed near its connection with the
main. This valve should be opened just enough to allow the steam-
pressure to act upon the air which occupies the space above the
water in the trap; but it should not be opened sufficiently to allow
the steam to enter in large volume and drive the air out. The
success of this arrangement depends upon keeping a layer or cushion
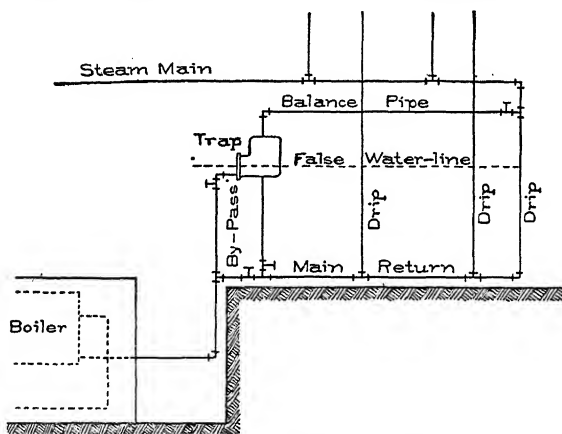
Fig. 79. Connections Made to Establish "False" Water-Line
when Boiler is below Basement Level

of cool air next to the surface of the water in the trap, and this is
easily done by following the method here described.

One-Pipe Relief System. In this system of piping, the radiators
have but a single connection, the steam flowing in and the condensa-
tion draining out through the same pipe. Fig. 80 shows the method
of running the pipes for this system. The steam main, as before,
leads from the top of the boiler, and is carried to as high a point as
the basement ceiling will allow; it then slopes downward with a
grade of about 1 inch in 10 feet, and makes a circuit of the building
or a portion of it.

Risers are taken from the top and carried to the radiators above,
as in the two-pipe system; but in this case, the condensation flows

back through the same pipe, and drains into the return main near the floor through drip connections which are made at frequent intervals. In a two-story building, the bottom of each riser to the second floor is dripped; and in larger buildings, it is customary to drip each riser that has more than one radiator connected with it. If the radiators are large and at a considerable distance from the next riser, it is better to make a drip connection for each
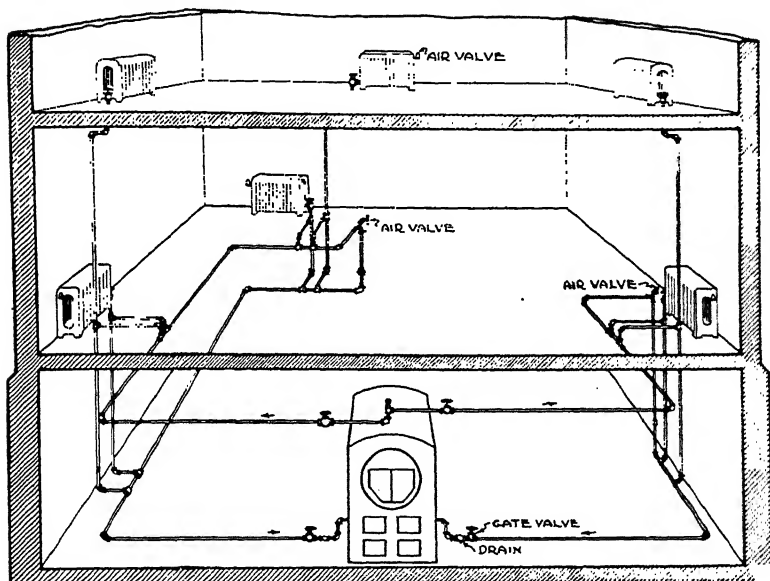
Fig. 80. Arrangement of Piping and Radiators in "One-Pipe Relief" System

radiator. When the return main is overhead, the risers should be dripped through siphon loops; but the ends of the branches should make direct connection with the returns. This is the reverse of the two-pipe system. In this case the lowest pressure is at the ends of the mains, so that steam introduced into the returns at these points will cause no trouble in the pipes connecting between these and the boiler.

If no steam is allowed to enter the returns, a vacuum will be formed, and there will be no pressure to force the water back to the boiler. A check-valve should always be placed in the main return near the boiler, to prevent the water from flowing out in case of a vacuum being formed suddenly in the pipes.

There is but little difference in the cost of the two systems, as larger pipes and valves are required for the single-pipe method. With radiators of medium size and properly proportioned connections, the single-pipe system is preferable, there being but one valve to operate and only one-half the number of risers passing through the lower rooms.

One-Pipe Circuit System. In this case, illustrated in Fig. 81, the steam main rises to the highest point of the basement, as before; and
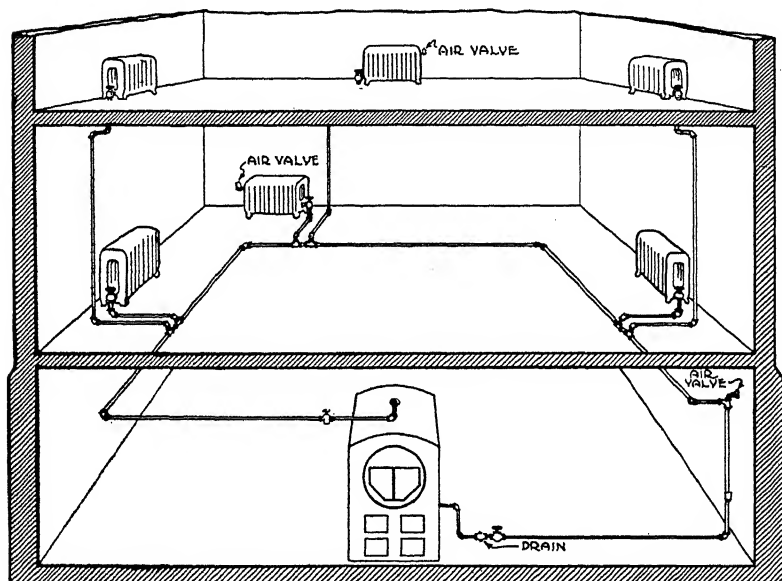
Fig. 81. Arrangement of Piping and Radiators in "One-Pipe Circuit" System

then, with a considerable pitch, makes an entire circuit of the building, and again connects with the boiler below the water-line. Single risers are taken from the top; and the condensation drains back through the same pipes, and is carried along with the flow of steam to the extreme end of the main, where it is returned to the boiler. The main is made large, and of the same size throughout its entire length. It must be given a good pitch to insure satisfactory results.

One objection to a single-pipe system is that the steam and return water are flowing in opposite directions, and the risers must be made

of extra large size to prevent any interference. This is overcome in large buildings by carrying a single riser to the attic, large enough to supply the entire building; then branching and running "drops" to the basement. In this system the flow of steam is downward, as well as that of water. This method of piping may be used with good results in two-pipe systems as well. Care must always be taken that no pockets or low points occur in any of the lines of pipe; but if for any reason they cannot be avoided, they should be carefully drained.

A modification of this system, adapting it to large buildings, is shown in diagram in Fig. 82. The riser shown in this case is one of several, the number depending upon the size of the building; and may be supplied at either bottom or top as most desirable. If steam is supplied at the bottom of the riser, as shown in the cut, all of the drip connections with the return drop, except the upper one, should
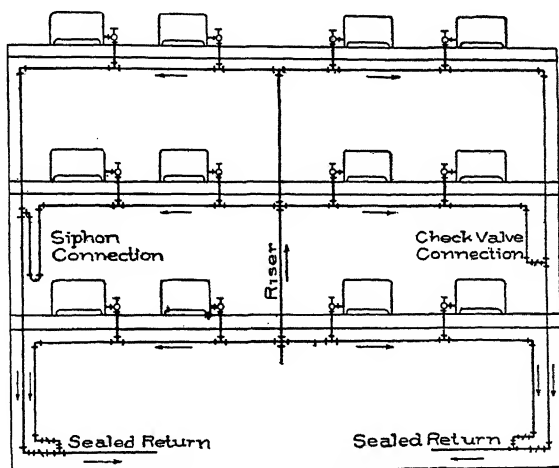
Fig. 82. "One-Pipe Circuit" System. Adapted to a Large Building

be sealed with either a siphon loop or a check-valve, to prevent the steam from short-circuiting and holding back the condensation in the returns above. If an overhead supply is used, the arrangement should be the reverse; that is, all return connections should be sealed except the lowest.

Sometimes a separate drip is carried down from each set of radiators, as shown on the lower story, being connected with the

main return below the water-line of the boiler. In case this is done, it is well to provide a check-valve in each drip below the water-line.

In buildings of any considerable size, it is well to divide the piping system into sections by means of valves placed in the corresponding supply and return branches. These are for use in case of a break in any part of the system, so that it will be necessary to shut off only a small part of the heating system during repairs. In tall buildings, it is customary to place valves at the top and bottom of each riser, for the same purpose.

Vapor Heating. Various types of vapor heating systems are in use for the heating of homes, schools, and public buildings, or any place where direct steam radiation may be used. A typical layout for such a system is shown in Fig. 83.

Such systems are of the two-pipe type using dry returns. The hot-water type of radiator is commonly used, having the sections connected by nipples at the top and bottom of the radiator.

The steam is supplied at the top tapping of the radiator through a graduated control valve by means of which variable quantities of steam may be admitted. The condensed steam and air from the radiator are discharged through a thermostatic trap located at the lower tapping on the opposite end of the radiator. This trap replaces the hand controlled outlet valve on the two-pipe system described before. The two-pipe system using a hand controlled outlet valve is seldom installed at the present time. The thermostatic traps used are similar to those used in the vacuum heating system.

The condensed steam and air from the radiators are carried back in the dry return piping toward the boiler. At the end of the dry return line is located a float vent valve which allows the air to be vented from the main to the basement, but which prevents the loss of steam or water from the dry returns. A direct return trap is installed as a safety device to return the condensed steam directly to the boiler when gravity return fails.

These systems operate with very low steam pressures and the temperature of the rooms may be more easily controlled by the amount of steam flowing into the radiators. Air valves on the radiators are eliminated so that there is no likelihood of foul air or water being discharged into the occupied rooms of the house.

BELLOWS
TRAP

FLOAT VENT VALVE
QUICK VENT VALVE
REGULATOR
COMPOUND GAUGE
QUICK VENT VALVE

DIRECT RETURN TRAP

SWING CHECKS

SCALE POCKET

BELLOWS
PACKLESS VALVES
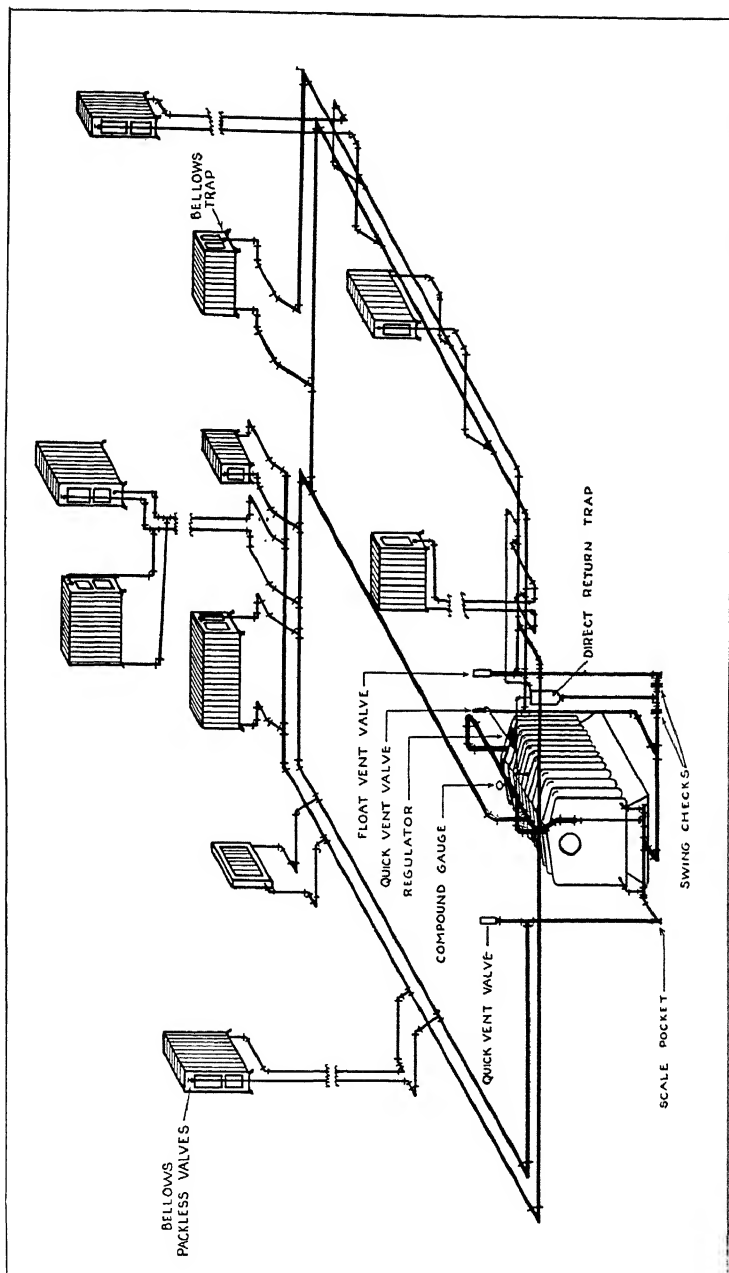
Fig. 83. The Trane Vapor Heating System

Radiator Connections. Figs. 84 and 85 show the common methods of making connections between supply pipes and radiators.
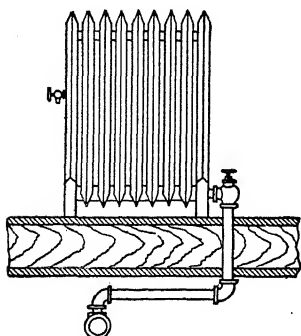
Fig. 84. "One-Pipe" Connection of Radiator
to Basement Main
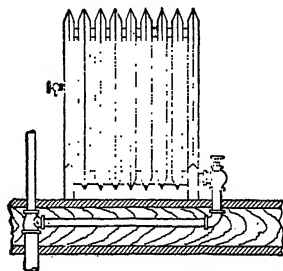
Fig. 85.   "One-Pipe" Connection
of Radiator to Riser

Fig. 84 shows a single-pipe connection with a basement main; and Fig. 85, a single-pipe connection with a riser.

Care must always be taken to make the horizontal part of the piping between the radiator and riser as short as possible and to give
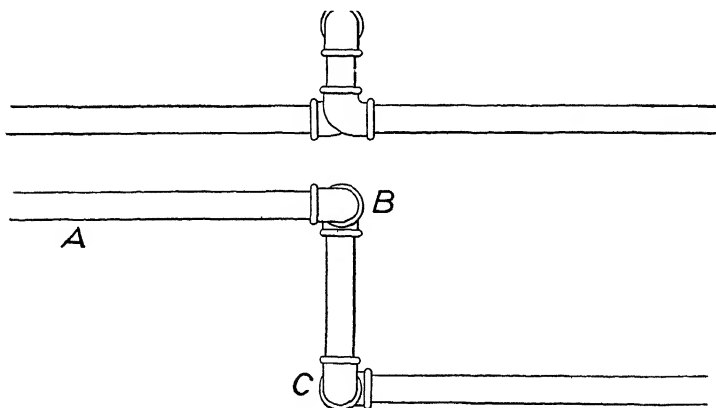
Fig. 86.   Elevation and Plan of Swivel-Joint to Counteract Effects of Expansion and
Contraction of Pipes

it a good pitch toward the riser. There are various ways of making these connections, especially suited to different conditions; but the

examples given serve to show the general principle to be followed.

Expansion of Pipes. Cold steam pipes expand approximately 1 inch in each 100 feet in length when low-pressure steam is turned into them. Therefore a system of piping must be planned to allow sufficient "spring" or "give" to the pipes to prevent injurious strains. This is done by means of offsets and bends. In the case of larger pipes this simple method will not be sufficient, and swivel or slip-joints must be used to take up the expansion.
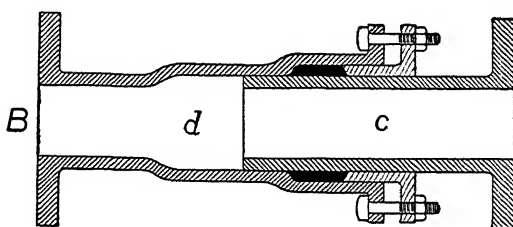
Fig. 87. "Slip-Joint" Connection to Take Care of Expansion
and Contraction of Pipes

The method of making up a swivel-joint is shown in Fig. 86. Any lengthening of the pipe A will be taken up by slight turning or swivel movements at the points B and C. A slip-joint is shown in Fig. 87. The part c slides inside the shell d and is made steam-tight by a stuffing-box, as shown. The pipes are connected at the flanges A and B.

When pipes pass through floors or partitions, the woodwork should be protected by galvanized-iron sleeves having a diameter from ¾ to 1 inch greater than the pipe. Fig. 88 shows a form of adjustable floor-sleeve which may be lengthened or shortened to conform to the thickness of floor or partition. If plain sleeves are used, a plate should be placed around the pipe where it passes through the floor or partition. These are made in two parts so they may be put in place after the pipe is hung. A plate of this kind is shown in Fig. 89.

Valves. The different styles commonly used for radiator connections are shown in Figs. 90, 91, and 92, and are known as *angle*, *offset*, and *corner* valves, respectively. An angle valve is used when the radiator is at the top of a riser or when the connections are like

those shown in Figs. 84 and 85; an offset valve is used when the connection between the riser and radiator is above the floor; and a corner valve is used when the radiator has to be set close in the corner of a room and there is not space for the usual connection.
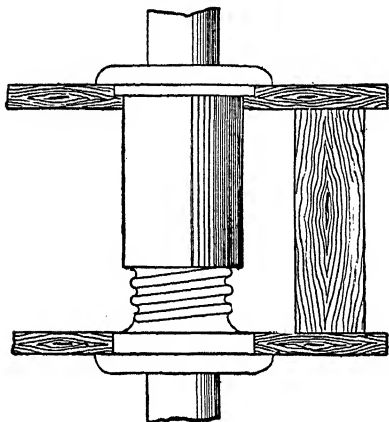
Fig. 88. Adjustable Metal Sleeve for Carrying Pipe through Floor or Partition
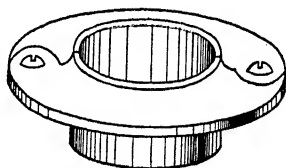
Fig. 89. Floor-Plate Adjusted to Plain Sleeve for Carrying Pipe through Floor or Partition

A globe valve never should be used in a horizontal steam supply or dry return. The reason for this is plainly shown in Fig. 93. In order for water to flow through the valve, it must rise to a height shown by the dotted line, which would half fill the pipes and cause

Fig. 90. Angle Valve
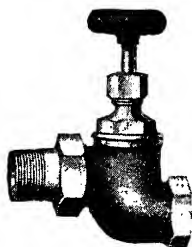
Fig. 91. Offset Valve
Valves for Radiator Connections
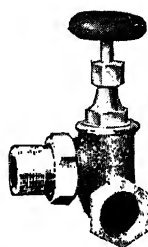
Fig. 92. Corner Valve

serious trouble from water-hammer. The gate valve shown in Fig. 94 does not have this undesirable feature, as the opening is on a level with the bottom of the pipe.

Air Valves. Valves of various kinds are used for freeing air from the radiators when steam is turned on. Fig. 95 shows a simple form of compression cock operated by hand. Fig. 96 is a type of
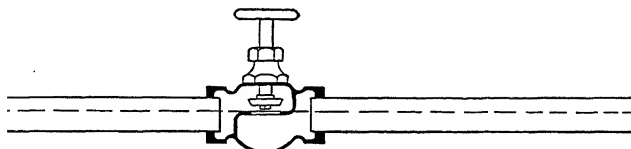
Fig. 93. Indicating Effect of Using Globe Valve on Horizontal Steam Supply Pipe or Dry Return
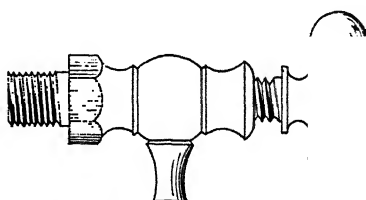
Fig. 95. Simplest Form of Air-Valve. Operated by Hand
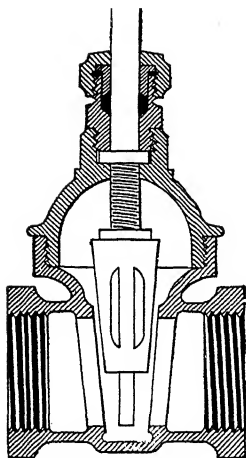
Fig. 94. Gate Valve
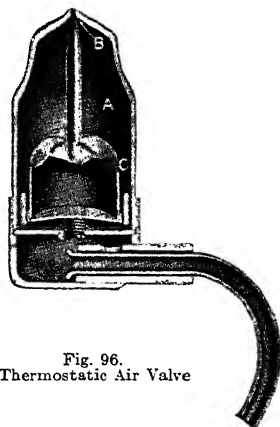
Fig. 96. Thermostatic Air Valve

automatic valve consisting of a shell A, which is attached to the radiator. B is a small opening which may be closed by the conical end of the spindle attached to the float C. Float C is partially filled with a volatile liquid.

When steam is turned into the radiator, the air is driven out at the opening B. As soon as steam surrounds the float the vaporization and expansion of the volatile liquid forces the metal diaphragm

of the float bottom downward against its support. This causes the float and its spindle to rise and close port B. When air collects in the valve, the volatile liquid cools and the air port B is opened. If the air valve is filled with water, the float is lifted by the water in the valve and the port B is closed, preventing the escape of water from the radiator. The curved siphon at the bottom of the valve is used to drain out the condensation of steam or the water from the valve shell after the air valve has been flooded.

Fig. 97. Expansion Chamber Air Valve
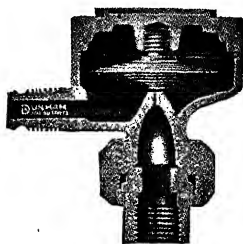
Fig. 98. Air Valve for Paul System

The valve shown in Fig. 97 is known as the expansion chamber type of valve and depends upon the expansion of the air in the outer annular chamber of the shell to force the water at the bottom of the shell to lift the float and close the valve. This valve also closes against the discharge of water.

An example of a Paul system air-line valve is shown in Fig. 98. This valve has a hollow diaphragm filled with a volatile liquid. Expansion or contraction of the liquid closes or opens the valve. The valve does not close against water.

A vacuum type of valve is shown in Fig. 99. The principle and action of the valve is similar to that of the valve shown in Fig. 96. Additional features are an air check at the air outlet 2, and a special spring bronze diaphragm 8, in the bottom of the outer shell. When the steam pressure is relieved on the radiator, the air check prevents air flowing in at the venting port. A partial vacuum is formed in the valve and the outside air pressure acting on diaphragm 8 forces the float and spindle up, closing the vent port.

VENTING PORT ② — ① AIR CHECK
FLOAT VALVE PIN ③

FLOAT ④ — ⑤ RIBS

FLEXIBLE DIAPHRAGM ⑦
VACUUM DIAPHRAGM ⑧
AIR CHANNEL
⑥ FLOAT SUPPORT
⑩ CHAMBER PORT
⑨ SIPHON
⑪ BASE
⑫ ATMOSPHERIC CHAMBER
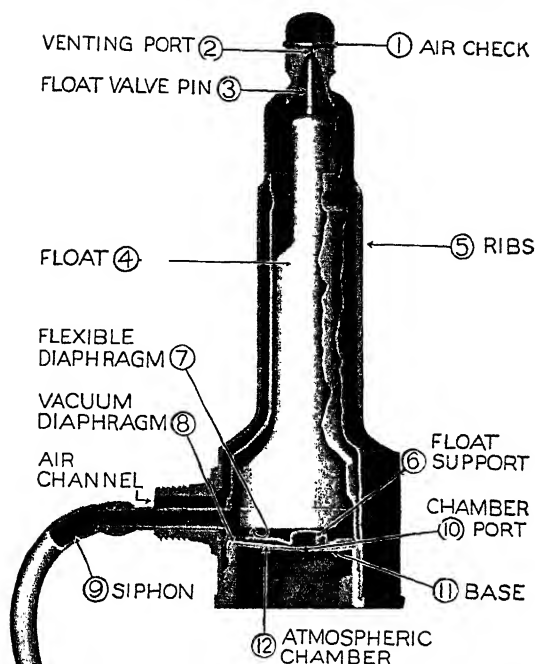
Fig. 99. Hoffman Siphon Air and Vacuum Valve

The advantage claimed for the vacuum valve is that the ordinary one-pipe steam heating system can be operated at a reduced steam temperature and pressure. This valve, to maintain a vacuum, must have the piping and radiator valves free from air leaks.

Pipe Sizes. The proportioning of the steam pipes in a heating plant is of the greatest importance, and should be carefully worked out by methods which experience has proved to be correct. There are several ways of doing this; but for ordinary conditions, Tables 51, 52, and 53 have given excellent results in actual practice. They have been computed from what is known as Babcock's formula, with suitable corrections made for actual working conditions. As the computations are somewhat complicated, only the results will be given here, with full directions for their proper use.

Table 51 gives the flow of steam in pounds per minute for pipes of different diameters and with one ounce drop in pressure between the supply and discharge ends of the pipe. These quantities are for pipes 100 feet in length; a pressure loss of one ounce per 100

Table 51. Flow of Steam in Pipes of Various Sizes and Equivalents to Be Added for Fittings

Pipe Lengths 100 Feet; Initial Steam Pressure 5 Pounds per Square Inch; Pressure Drop 1 Ounce

Size of Pipe, Inches	Flow of Pounds per Minute	Equivalent Length to Be Added for Each Globe Valve and Entrance Feet	Equivalent Length to Be Added for Each 90 Degree Elbow, Feet	Size of Pipe, Inches	Flow of Pounds per Minute	Equivalent Length to Be Added for Each Globe Valve and Entrance Feet	Equivalent Length to Be Added for Each 90 Deg Elbow,]
1	0.256	2	1.5	6	34.3	37	24
1¼	0.568	4	3.0	7	52.8	44	29
1½	0.886	5	3.5	8	70.8	53	35
2	1.745	7	5	9	99.4	61	41
2½	2.92	10	6	10	130.5	70	47
3	5.28	14	9	11	167.0	78	52
3½	8.07	17	11	12	208.0	86	58
4	11.3	20	13	14	314.5	106	70
5	20.9	28	19				

Table 52. Factors for Calculating Flow of Steam in Pipes Under Initial Pressures Other Than Five Pounds
(To be used in connection with Table 51)

Initial Pressure Pounds per Square Inch							
2	10	20	30	40	50	60	80
0.92	1.11	1.33	1.46	1.6	1.73	1.85	2.07

feet equivalent length of pipe is satisfactory in heating work. As the length of pipe increases, friction becomes greater, and the loss in the steam pressure increases.

When calculations of the flow of steam are made, it is always necessary to consider the effect of valves, elbows, and other obstructions in the pipe line. The usual procedure is to estimate the resistance effect of these fittings in linear feet of straight pipe in the steam line under consideration. The frictional effect of various fittings for different pipe sizes is included in Table 51. When the

Table 53. Total Pressure Loss, Ounces. Pipes of Various Lengths.
Pressure Loss 1 Ounce per 100 Feet

Feet	Pressure Loss Ounces	Feet	Pressure Loss Ounces	Feet	Pressure Loss Ounces	Feet	Pressure Loss Ounces
10	0.1	120	1.2	275	2.75	600	6.0
20	0.2	130	1.3	300	3.0	650	6.5
30	0.3	140	1.4	325	3.25	700	7.0
40	0.4	150	1.5	350	3.5	750	7.5
50	0.5	160	1.6	375	3.75	800	8.0
60	0.6	170	1.7	400	4.0	850	8.5
70	0.7	180	1.8	425	4.25	900	9.0
80	0.8	190	1.9	450	4.5	950	9.5
90	0.9	200	2.0	475	4.75	1,000	10.0
100	1.0	225	2.25	500	5.0		
110	1.1	250	2.5	550	5.5		

allowances for valves, elbows, etc., are added to the length of straight pipe, the sum is spoken of as the total equivalent length of the pipe. The total equivalent length should always be used in the calculation of the steam capacity of a pipe line.

If the drop in pressure is less than 1 ounce, the actual discharge will be slightly less than the quantities given in the table. However, this difference will be small for pressures up to 5 pounds, and may be neglected. For other initial pressures, Table 52 has been prepared. This is to be used in connection with Table 51 as follows: First find from Table 51 the quantity of steam which will be discharged through the given diameter of pipe, then look in Table 52 for the factor corresponding with the higher initial pressure to be used. The quantity given in Table 51, *multiplied by* this factor, will give the actual capacity of pipe under the given conditions.

Example 1. What weight of steam will be discharged through a 3-inch pipe 100 equivalent feet long, with an initial pressure of 60 pounds?

Table 51 indicates that a 3-inch pipe will discharge 5.28 pounds of steam per minute with 1 ounce pressure drop when the initial pressure is 5 pounds per square inch. The factor corresponding to 60 pounds initial pressure is given as 1.85 in Table 52. According to the rule given, $5.28 \times 1.85 = 9.8$ pounds, which is the capacity of a 3-inch pipe under the assumed conditions.

Example 2. What quantity of steam, at 80 pounds per square inch, may be discharged through a 6-inch boiler outlet pipe 79 feet actual length? The pipe has one globe valve and three 90° elbows.

The total equivalent length of 6-inch pipe, using the allowances given for fittings in Table 51, is computed as shown below.

Entrance loss equivalent at the boiler steam nozzle	= 37 feet
Globe valve equivalent	= 37 feet
Equivalent of three 90° elbows, 3×24	= 72 feet
Length of straight pipe	= 79 feet
Total equivalent length	= 225 feet

Pressure loss of steam = $(225 \times 1) \div 100 = 2.25$ ounces as in Table 53.

Flow of steam per minute at 5 pounds per square inch initial pressure, from Table 51	= 34.3 pounds
Factor for 80 pounds per square inch initial pressure, Table 52	= 2.07
Steam discharge per minute = 34.3×2.07	= 71 pounds

Example 3. What size of pipe will be necessary for a boiler to deliver 90 pounds of steam per minute at 50 pounds pressure 200 feet away? There are two globe valves and five 90° elbows in the line. Find the pressure loss in the steam.

Factor for initial pressure of 50 pounds per square inch
$$\text{Table 52} = 1.73$$
Pipe capacity required at 5 pounds per square inch
$$= 90 \div 1.73 = 52 \text{ pounds per minute}$$

From Table 51, select a 7-inch pipe.
Total equivalent length of 7-inch pipe is computed as

Length of straight pipe	$= 200$ feet
Entrance equivalent	$= 44$
Globe valves	$= 2 \times 44 = 88$
Elbows	$= 5 \times 29 = 145$
	477 feet

$$\text{Pressure loss} = (477 \times 1) \div 100 = 4.77 \text{ ounces.}$$

Table 54. Heating Surface Supplied by Pipes of Various Sizes
Length of Pipe, 100 Feet

Size of Pipe Inches	Square Feet of Heating Surface	Size of Pipe Inches	Square Feet of Heating Surface	Size of Pipe Inches	Square Feet of Heating Surface
1	50	2½	580	5	4,180
1¼	110	3	1,050	6	6,860
1½	175	3½	1,610	7	10,560
2	350	4	2,260	8	14,160

In computing the pipe sizes for a heating system by the above methods it would be a long process to work out the size of each branch separately. Accordingly Table 54 has been prepared for ready use in low-pressure work.

As most direct heating systems, and especially those in schoolhouses, are made up of both radiators and circulation coils, an efficiency of 250 B.t.u.'s has been taken for direct radiation of whatever variety, no distinction being made between the different kinds. This gives a slightly larger pipe than is necessary for cast-iron radiators; but it is probably offset by bends in the pipes, and in any case gives a slight factor of safety. We find from a steam table that the *latent heat* of steam at 20 pounds above a vacuum (which corresponds to 5 pounds' gauge-pressure) is $962+$ B.t.u.—which means that for every pound of steam condensed in a radiator, 962 B.t.u.'s are given off for warming the air of the room. If a radiator has an efficiency of 250 B.t.u.'s, each square foot of surface will condense $250 \div 962 = 0.26$ pound of steam per hour. Then we may assume in round numbers a condensation of ⅓ of a pound of steam per hour for each square foot of direct radiation, when computing the sizes of steam pipes in low-pressure heating. Table 54 has been calculated

Table 55. Radiating Surface Supplied by Steam Risers		Table 56. Radiating Surface Supplied by Steam Risers	
10 Feet per Second Velocity		15 Feet per Second Velocity	
Size of Pipe Inches	Radiation Square Feet	Size of Pipe Inches	Radiation Square Feet
1	30	1	50
1¼	60	1¼	90
1½	80	1½	120
2	130	2	200
2½	190	2½	290
3	290	3	340
3½	390	3½	590

on this assumption and gives the square feet of heating surface which different sizes of pipe will supply, with a drop in pressure of one ounce in each 100 feet of pipe. The sizes of long mains and special pipes of large size should be proportioned directly from Tables 51, 52, and 53.

Where the two-pipe system is used and the radiators have separate supply and return pipes, the risers or vertical pipes may be taken from Table 54; but if the single-pipe system is used, the risers must be increased in size, as the steam and water are flowing in opposite directions and must have plenty of room to pass each other. It is customary in this case to base the computation on the velocity of the steam in the pipes rather than on the drop in pressure.

Assuming, as before, a condensation of one-third of a pound of steam per hour per square foot of radiation, Tables 55 and 56 have been prepared for velocities of 10 and 15 feet per second. The sizes given in Table 56 have been found sufficient in most cases; but the larger sizes, based on a flow of 10 feet per second, give greater safety and should be more generally used.

The size of the largest riser should usually be limited to 2½ inches in school and dwelling-house work, unless it is a special pipe carried up in a concealed position. If the length of riser is short between the lowest radiator and the main, a higher velocity of 20 feet or more may be allowed through this portion instead of making the pipe excessively large.

Pipes running long distances through partitions and between floors should, if possible, be designed so that the pipes plus pipe covering can be wholly concealed. Pipe covering should be used to prevent some pipe losses and also to prevent the unnecessary appli-

Table 57. Sizes of Returns for Steam Pipes

Diam. of Steam Pipe Inches	Diam. of Dry Return Inches	Diam. of Sealed Return Inches	Diam. of Steam Pipe Inches	Diam. of Dry Return Inches	Diam. of Sealed Return Inches	Diam. of Steam Pipe Inches	Diam. of Dry Return Inches	Diam. of Sealed Return Inches
1		¾	3		2	7		
1¼			3½		2	8		
1½			4		2½	9		
2			5		2½	10		
2½			6		3	12		

cation of heat to plaster and flooring. Pipe covering is generally of 1-inch, 1½-inch, 2-inch, or 3-inch thickness according to the necessity. Manufacturers' catalogues should be consulted to insure accurate results.

Returns. The size of return pipes is usually a matter of custom and judgment rather than computation. It is a common rule among steamfitters to make the returns one size smaller than the corresponding steam pipes. This is a good rule for the smaller sizes, but gives a larger return than is necessary for the larger sizes of pipe. Table 57 gives different sizes of steam pipes with the corresponding diameters for dry and sealed returns.

The length of run and number of turns in a return pipe should be noted, and provision made for any unusual conditions. Where the condensation is discharged through a trap into a lower pressure, the sizes given may be slightly reduced, especially among the larger sizes, depending upon the differences in pressure.

Table 58. Pipe Sizes for Radiator Connections

Square Feet of Radiation		Steam Pipe Inches	Return Pipe Inches	Square Feet of Radiation		Steam Pipe Inches	Return Pipe Inches
Two-Pipe	0 to 30	¾	¾	Single-Pipe	0 to 24	1	...
	30 to 48	1	¾		24 to 60	1¼	...
	48 to 96	1¼	1		60 to 80	1½	...
	96 to 150	1½	1¼		80 to 130	2	...

Table 59. Pipe Sizes from Boiler to Main Header

Diameter of Boiler Inches	Size of Steam Pipe Inches	Diameter of Boiler Inches	Size of Steam Pipe Inches
36	3	60	5
42	4	66	6
48	4	72	6
54	5	..	...

Radiators are usually tapped for pipe connections as shown in Table 58, and these sizes may be used for the connections with the mains or risers.

Boiler Connections. The steam main should be connected to the rear nozzle, if a tubular boiler is used, as the boiling of the water is
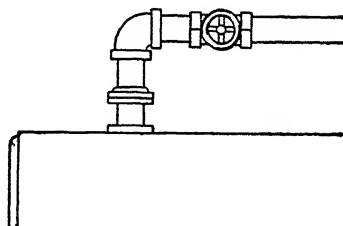
Fig. 100. Good Position for Shut-Off Valve

less violent at this point and dryer steam will be obtained. The shut-off valve should be placed in such a position that pockets for the accumulation of condensation will be avoided. Fig. 100 shows a good position for the valve.

The size of steam connection may be computed by means of the methods already given, if desired. But for convenience the sizes given in Table 59 may be used with satisfactory results for the short runs between the boilers and main header.

The return connection is made through the blow-off pipe, and should be arranged so that the boiler can be blown off without draining the returns. A check-valve should be placed in the main
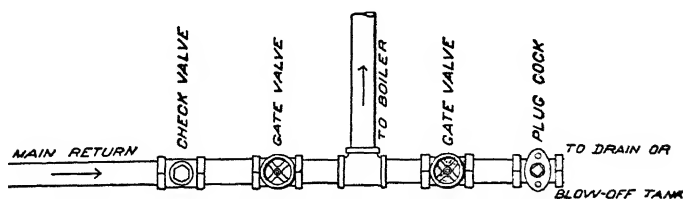
Fig. 101. A Good Arrangement of Return and Blow-Off Connections

return, and a plug-cock in the blow-off pipe. Fig. 101 shows in plan a good arrangement for these connections.

In the best class of work the feed connections (with the exception of that part exposed in the smoke-bonnet) are always made of brass. The small section referred to should be of extra heavy wrought

Table 60. Sizes for Return, Blow=Off, and Feed Pipes

Diameter of Boiler Inches	Size of Pipe for Gravity Return Inches	Size of Blow-off Pipe Inches	Size of Feed Pipe Inches	Diameter of Boiler Inches	Size of Pipe for Gravity Return Inches	Size of Blow-off Pipe Inches	Size of Feed Pipe Inches
36	1½	1¼	1	60	2½	2	1¼
42	2	1½	1	66	3	2½	1½
48	2	1½	1	72	3	2½	1½
54	2½	2	1¼	...	...	...	...

iron. The branch to each boiler should be provided with a gate or globe valve and a check-valve, the former being placed next to the boiler.

Table 60 gives suitable sizes for return, blow-off, and feed pipes for boilers of different diameters.

Blow=Off Tank. Where the blow-off pipe connects with a sewer, some means must be provided for cooling the water, or the
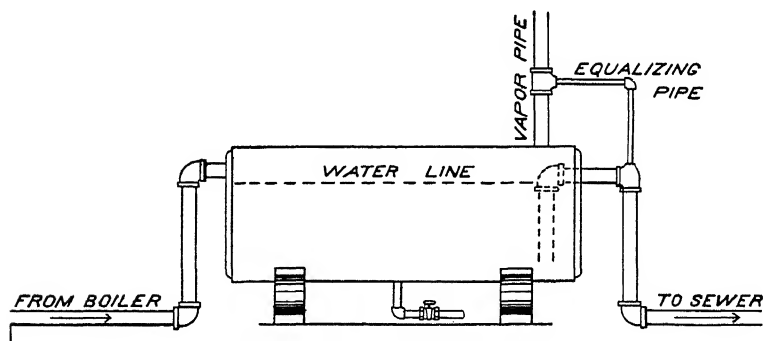
Fig. 102. Connections of Blow-Off Tank

expansion and contraction caused by the hot water flowing through the drain-pipes will start the joints and cause leaks. For this reason it is customary to pass the water through a blow-off tank. A form of wrought-iron tank is shown in Fig. 102. It consists of a receiver supported on cast-iron cradles. The tank ordinarily stands nearly full of cold water.

The pipe from the boiler enters above the water-line, and the sewer connection leads from near the bottom, as shown. A vapor pipe is carried from the top of the tank above the roof of the building. When water from the boiler is blown into the tank, cold water from the bottom flows into the sewer, and the steam is carried off through

the vapor pipe. The equalizing pipe is to prevent any siphon action which might draw the water out of the tank after a flow is once started. As only a part of the water is blown out of a boiler at one time, the blow-off tank can be of a comparatively small size. A tank 24 by 48 inches should be large enough for boilers up to 48 inches in diameter; and one 36 by 72 inches should care for a boiler 72 inches in diameter. If smaller quantities of water are blown off at one time, smaller tanks can be used. The sizes given in Table 60 are sufficient for batteries of two or more boilers, as one boiler can be blown off and the water allowed to cool before a second one is blown off. Cast-iron tanks are often used in place of wrought-iron, and these may be sunk in the ground if desired.

PRACTICE PROBLEMS

1. In the section on Radiators, Problems 1 and 2 on pages 163 and 164 concern the house shown in Figs. 69 and 70. Assume the radiator sizes calculated in those problems. Design a one-pipe circuit system for this house. Illustrate with a drawing in either a plan view or a sketch similar to the figures in this section. Indicate all places where pipe slant is required. Indicate all air vents.

2. Determine the practical size for all pipes shown in the answer to Problem 1.

3. Refer to Problem 3, page 94, in the section on Boilers. Design a two-pipe steam system for the residence shown in Figs. 27, 28, and 29. Illustrate with a drawing in either a plan view or a sketch similar to the figures in this section. Indicate all places where pipe slant is required. Indicate all air vents.

4. Determine the practical size for all pipes shown in the answer to Problem 3.
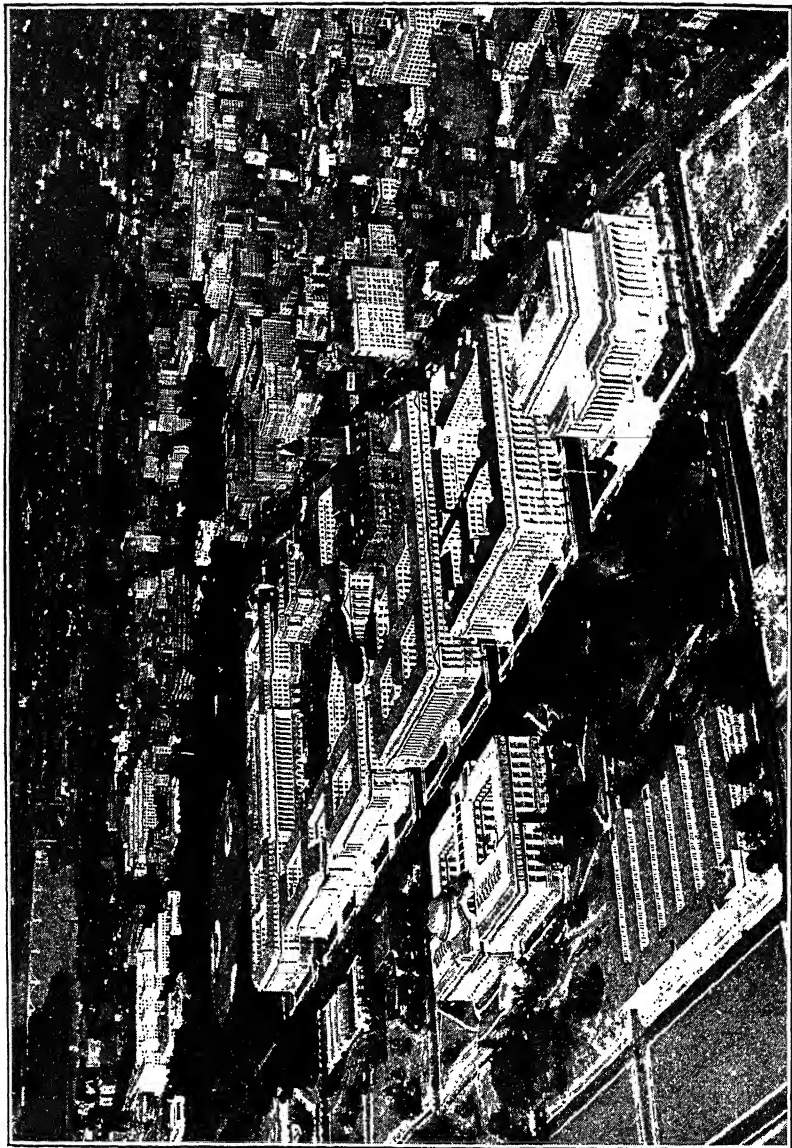
NEW FEDERAL TRIANGLE, WASHINGTON, D. C., SHOWING TYPES OF BUILDINGS BEING AIR CONDITIONED

Starting with the Archives Building at the point of the triangle to the wide spread of the Department of Commerce at the wide end, it shows the Interstate Commerce Building, Justice Building and the circular court of the Post Office Department. In the distance on the left may be seen the Interior Building.

Photo by Underwood & Underwood

CHAPTER X

*HOT=WATER HEATING

Types of Heaters. Hot-water heaters differ from steam boilers principally in the omission of the reservoir or space for steam above the heating surface. The steam boiler might answer as a heater for hot water; but the large capacity left for the steam would tend to make its operation slow and rather unsatisfactory, although the same type of boiler is sometimes used for both steam and hot water. The passages in a hot-water heater need not extend so directly from bottom to top as in a steam boiler, since the problem of providing for the free liberation of the steam bubbles does not have to be considered. In general, the heat from the furnace should strike the surfaces in such a manner as to increase the natural circulation. This may be accomplished to a certain extent by arranging the heating surface so a large proportion of the direct heat will be absorbed near the top of the heater. Practically the boilers for low-pressure steam and for hot water differ from each other very little as to the character of the heating surface, so that the methods given for computing the size of grate surface, horsepower, etc., for steam boilers can be used.

It is sometimes stated that, owing to the greater difference in temperature between the furnace gases and the water in a hot-water heater, as compared with steam, the heating surface will be more efficient and a smaller heater can be used. While this is true, authorities agree that this advantage is so small that no account should be taken of it, and the general proportions of the heater should be calculated in the same manner as for steam. Fig. 103 shows a form of heater made up of slabs or sections similar to the sectional steam boiler. The size can be increased in a similar manner by adding more sections. In this case, however, the boiler is increased in width instead of in length. This has an advantage in the larger sizes, as a second fire door can be added, and all parts of the grate can be reached as well in the large sizes as in the small.

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*In the chapter on "Boilers" are shown the new types of boilers used for either hot water or steam. The information on the old style boilers is given here to aid those readers who may encounter them at one time or another. To figure sizes of modern hot water radiators, use the method explained on page 150. Data relative to radiators (steam, vapor, or hot water) see pages 141–169.
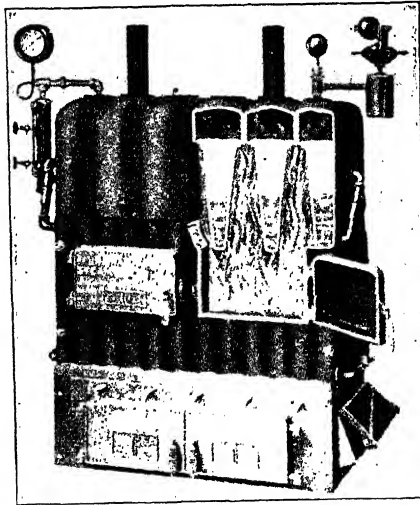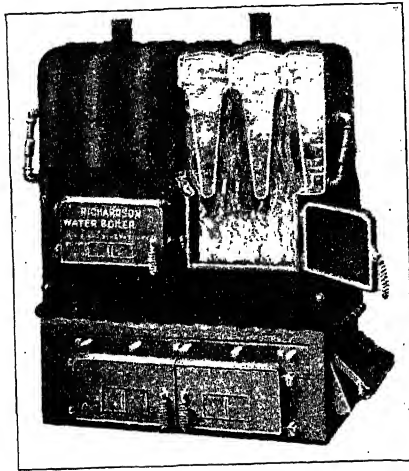
Fig. 103. Top—Richardson Sectional Hot-Water Heater.
Bottom—Same Heater Equipped as Steam Boiler

Fig. 104 shows another type of sectional cast-iron heater. A deep fire chamber with corrugated sides makes this furnace a quick heater and keeps the fire a long time without attention. The space between the outer and inner corrugated shells surrounding the furnace is filled with water, as is also the case with the cross-pipes directly over the fire and the drum at the top.

The ordinary horizontal and vertical tubular boilers, with various modifications, are used to a considerable extent for hot-water heating and are well adapted to this class of work, especially in the case of large buildings.

Automatic regulators are used for the purpose of maintaining a constant temperature of the water leaving the heater. A common and much used type of regulator is illustrated by Fig. 105. This
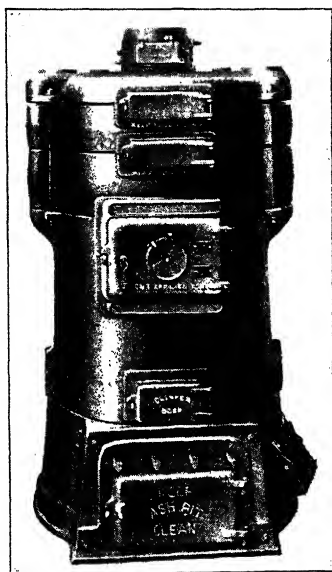
Fig. 104. Cast-Iron Heater
Courtesy of Burham Boiler Company

regulator is made entirely of metal and consists of an outer shell, a metallic sylphon bellows, a stem, lever, and counterweights. The stem is attached within and at the bottom of the sylphon bellows and passes out of the top part of the shell where it comes in contact with the lever through a knife edge.

The upper part of the sylphon bellows is flanged and the flange rests between the upper and lower parts of the shell. This construction makes a packless joint, which in all respects is considered more desirable. The lower part of the shell is filled with a volatile liquid which surrounds the sylphon bellows.

When the volatile liquid is heated, it vaporizes and a pressure is exerted upon the lower part of the sylphon bellows. The pressure tends to contract the bellows and the stem is moved upward, thus moving the lever carrying the counterweights. The movement of the lever may be transmitted by chains to open and close the water heater dampers. Adjustments of the counterweights regulate the
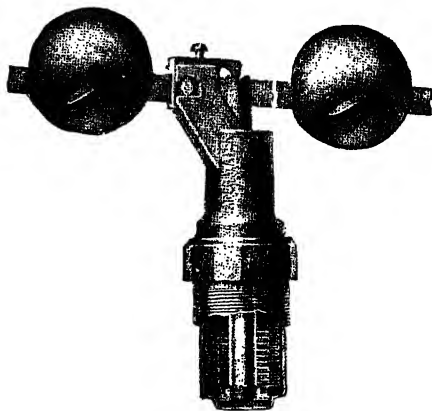
Fig. 105. Hot-Water Heater Regulator
Courtesy of American Radiator Company

temperature required to create pressure enough within the shell to cause the lever to close or open the dampers. These regulators are placed in the outlet pipe just above the top of the heater and in an accessible position.

A hot-water system is similar in construction and operation to one designed for steam, except that hot water flows through the pipes and radiators.

Both low and high temperature water are used. The low temperature range is thought of as that which affords a heat emission of 150 to 160 B.t.u.'s per square foot of radiation, while the high temperature range delivers 200 to 240 B.t.u.'s per square foot of radiation. An advantage of the high temperature range is that it permits the use of the smaller, more adaptable radiators.

The circulation through the pipes is produced solely by the difference in weight of the water in the supply and return, due to the difference in temperature. When water is heated, it expands

and thus a given volume becomes lighter and tends to rise, and the cooler water flows in to take its place; if the application of heat is kept up, the circulation thus produced is continuous. The velocity of flow depends upon the difference in temperature between the supply and return and the height of the radiator above the boiler. The horizontal distance of the radiator from the boiler is also an important factor affecting the velocity of flow.

This action is best shown by means of a diagram, Fig. 106. If a glass tube of the form shown in the figure is filled with water and held in a vertical position, no movement of the water will be noticed, because the two columns A and B are of the same weight, and therefore in equilibrium. Now, if a lamp flame be held near the tube A, the small bubbles of steam which are formed will show the water
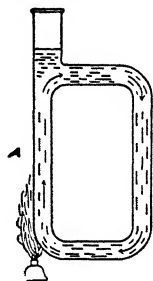
Fig. 106.   Illustrating How the Heating
of Water Causes Circulation

to be in motion, with a current flowing in the direction indicated by the arrows. The reason for this is, that, as the water in A is heated it expands and becomes lighter for a given volume and is forced upwards by the heavier water in B falling to the bottom of the tube. The heated water flows from A through the connecting tube at the top into B, where it takes the place of the cooler water which is settling to the bottom. If, now, the lamp be replaced by a furnace, and the columns A and B be connected at the top by inserting a radiator, the illustration will assume the practical form as utilized in hot-water heating, Fig. 107.

The heat given off by the radiator always insures a difference in temperature between the columns of water in the supply and return pipes, so that as long as heat is supplied by the furnace the flow of water will continue. The greater the difference in

temperature of the water in the two pipes, the greater the difference
in weight, and consequently the faster the flow. The greater the
height of the radiator above the heater, the more rapid will be the
circulation, because the total difference in weight between the water
in the supply and return risers will vary directly with their height.
From the above it is evident that the rapidity of flow depends
chiefly upon the temperature difference between the supply and
return, and upon the height of the radiator above the heater.   An-
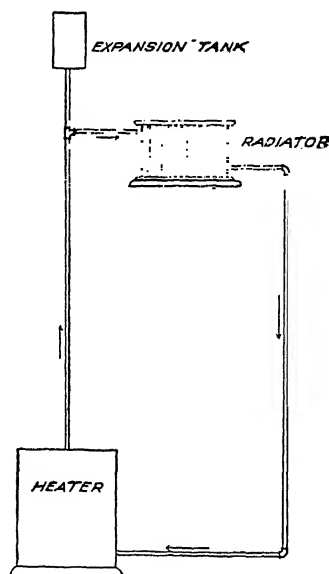
Fig. 107.   Illustrating Simple Circulation in a Heating System

other factor which must be considered in long runs of horizontal
pipe is the frictional resistance.

Systems of Circulation.   There are two distinct systems of cir-
culation employed—one depending on the difference in temperature
of the water in the supply and return pipes, called *gravity circulation;*
and another where a pump is used to force the water through the
mains, called *forced circulation.*   The former is used for dwellings
and other buildings of ordinary size, and the latter for large buildings
and especially where there are long horizontal runs of pipe.

For gravity circulation some form of sectional cast-iron boiler
is commonly used, although steel tubular boilers may be employed

if·desired. In the case of forced circulation, a heater designed to warm the water by means of live or exhaust steam is often used. A centrifugal or rotary pump is best adapted to this purpose, and it may be driven by an electric motor or a steam engine, as most convenient.

Systems of Piping. A system of hot-water heating should produce a perfect circulation of water from the heater to the radiating
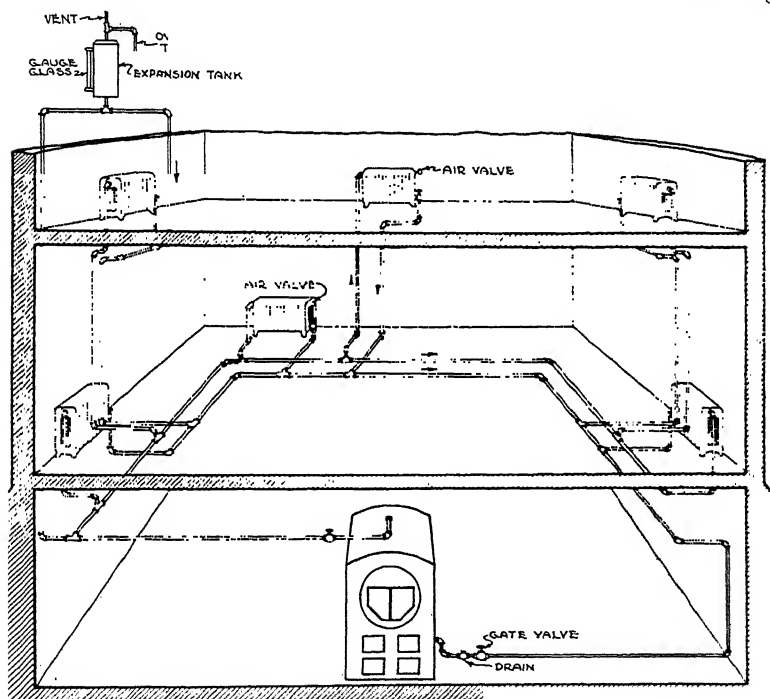
Fig. 108. Two-Pipe Up-Feed, Reversed Return, Hot-Water Heating System

surface, and thence back to the heater through the returns. A satisfactory system of piping employed for hot-water heating is shown in Fig. 108. In this arrangement the supply main and branches have an inclination upward from the heater; the returns are parallel to the mains, and have an inclination downward toward the heater, connecting with it at the lowest point. The flow pipes or risers are taken from the tops of the mains, and may supply one or more radiators as required. The return risers or drops are connected into the return mains at the side.

It will always be found that the principal current of heated water will take the path of least resistance, and that a small obstruction or irregularity in the piping is sufficient to interfere greatly with amount of heat received in different parts of the same system.

Oftentimes hot-water systems are installed so that the water in the supply main and the water in the return main flow in opposite
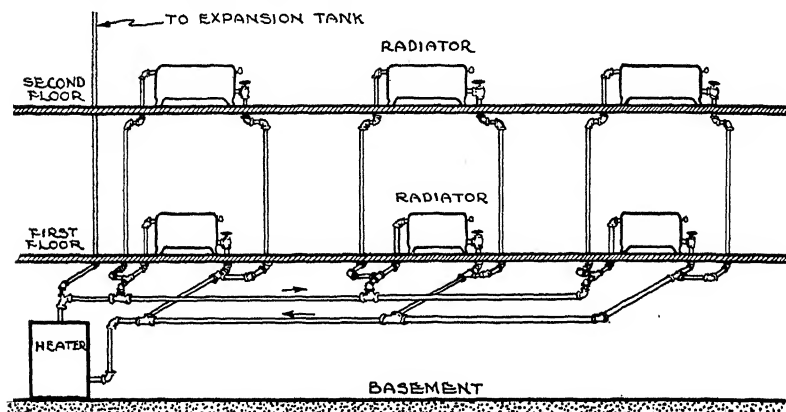
Fig. 109. Two-Pipe Up-Feed, Short-Circuited Return, Hot-Water Heating System

directions. As the flow of water takes place through the part of the circuit with the least resistance, the radiators near the heater work well, while those farther away are likely to fail to heat. This is especially true in the two-pipe up-feed system shown in Fig. 109.

The arrangement shown in Fig. 110 is similar to the circuit system for steam, except that the radiators have two connections instead of one. This method is especially adapted to apartment houses, where each apartment has its separate heater, as it eliminates a separate return main and thus reduces, by practically one-half, the amount of piping in the basement. The supply risers are taken from the top of the main; while the returns should connect into the side or bottom a short distance beyond, and in a direction away from the boiler. When this system is used, it is necessary to enlarge the radiators slightly as the distance from the boiler increases.

In apartments of eight or ten rooms, the size of the last radiator may be increased from 10 to 15 per cent, and the intermediate ones proportionally, at the same time keeping the main of a large and uniform size for the entire circuit.

Attention is called to the method of piping the inlet and outlet connections of the hot-water radiators in Fig. 110. Oftentimes the hot water is fed into the radiator at the bottom and the return water taken out at the bottom at the opposite end. When both piping connections to the radiator are made at the bottom the radiator cannot be positively controlled except by the use of two valves, one at the inlet end and one at the outlet end of the radiator.

Positive control of a hot-water radiator can be secured by the use of one valve at the outlet end if the supply line enters the radiator at the top tapping as shown in Figs. 108, 109, and 111. Furthermore,
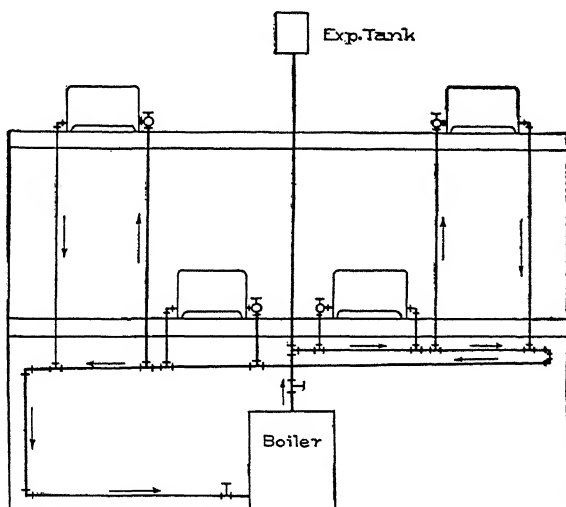
Fig. 110. One Main Hot-Water Heating System

tests have shown that a radiator will emit more heat when the supply connection is at the top and the return connection is at the bottom of the radiator, other conditions being the same.

Overhead Distribution.   This system of piping is shown in Fig. 111. A single riser is carried directly to the attic. Branches are taken from attic mains to supply the various drops to which the radiators are connected. An important advantage in connection with this system is that the air rises at once to the expansion tank and escapes through the vent, so that air-valves are not required on the radiators.

At the same time, it has the disadvantage that the water in the tank is under less pressure than in the heater; hence, it will boil at a lower temperature. No trouble will be experienced from this however, unless the temperature of the water is raised above 212 degrees.

Expansion Tank. Every system for hot-water heating should be connected with an expansion tank placed at a point somewhat above the highest radiator. The tank must in every case be connected to a line of piping which cannot by any possible means be shut off from the boiler. When water is heated, it expands a certain amount, depending upon the temperature to which it is raised; and a tank or reservoir should always be provided to care for this increase in volume.
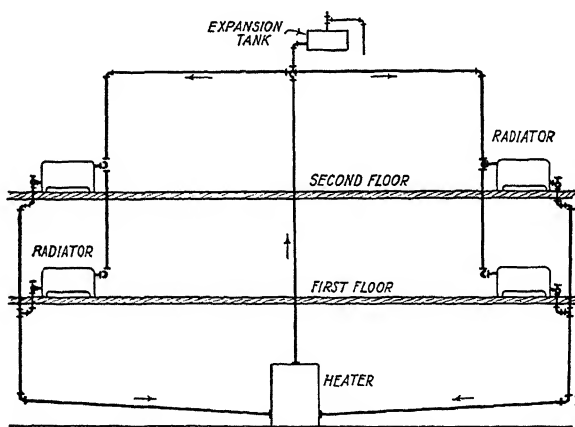
Fig. 111. Overhead Distribution System of Hot-Water Piping

Expansion tanks are usually made of heavy galvanized iron of one of the forms shown in Figs. 112 and 113, the latter form being used where the headroom is limited. The connection from the heating system enters the bottom of the tank, and an open vent pipe is taken from the top. An overflow connected with a sink or drain-pipe should be provided. A ball-cock is sometimes arranged to keep the water-line in the tank at a constant level.

An altitude gauge is often placed in the basement with the colored hand or pointer set to indicate the normal water-line in the expansion tank. When the movable hand falls below the fixed one, more water may be added, as required, through the supply pipe at

the boiler. When the tank is placed in an attic or roof space where there is danger of freezing, the expansion pipe may be connected 6 or 8 inches from the bottom, and a circulation pipe connected
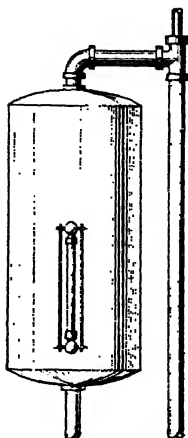

Fig. 112. Common Form of Galvanized-Iron
Expansion Tank

with the return from an upper-floor radiator as shown in Fig. 108. This produces a slow circulation near the tank and keeps the water warm.
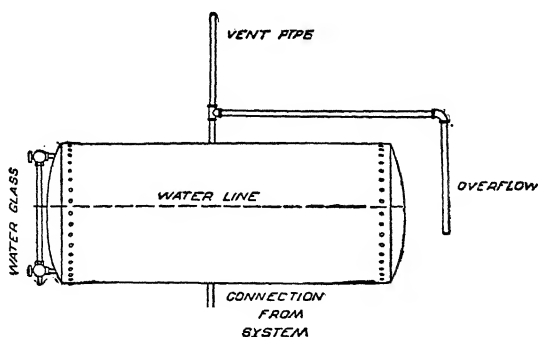

Fig. 113. Form of Expansion Tank Used where Headroom Is Limited

The size of the expansion tank depends upon the volume of water contained in the system, and on the temperature to which it is heated. The following rule for computing the capacity of the tank may be used with satisfactory results:

Square feet of radiation divided by 40 equals required capacity of tank in gallons.

Air=Venting. One very important point to be kept in mind in the design of a hot-water system is the removal of air from the pipes and radiators. When the water in the boiler is heated, the air it contains forms into small bubbles which rise to the highest points of the system.

In the arrangement shown in Fig. 109, the main and branches grade upward from the boiler so that the air finds its way into the radiators, from which it may be drawn off by means of the air-valves.

A better plan is the one shown in Fig. 110. In this case the expansion pipe is taken directly off the top of the main over the boiler so that the larger part of the air rises directly to the expansion tank and escapes through the vent pipe. The same action takes place in the overhead system shown in Fig. 111, where the top of the main riser is connected with the tank. Every high point in the system and every radiator, except in the downward system with top supply connection, should be provided with an air-valve.

Pipe Connections. There are various methods of connecting the radiators with the mains and risers. Fig. 114 shows a radiator connected with the horizontal flow and return mains, which are located below the floor. The manner of connecting with a vertical riser and return drop is shown in Fig. 115. As the water tends to flow to the highest point, the radiators on the lower floors should be favored by making the connection at the top of the riser and taking the pipe for the upper floors from the side as shown. Fig. 116 illustrates the manner of connecting with a radiator on an upper floor where the supply is connected at the top of the radiator. The method of locating the supply and return connections shown in Figs. 114 and 115 does not insure positive control of the radiators. The supply and return connections should always be made at the radiators as shown in Figs. 116, 117, and 118.

The connections shown in Figs. 117 and 118 are used with the overhead system shown in Fig. 111.

Where the connection is of the form shown in Fig. 117, the cooler water from the radiators is discharged into the supply pipe again so that the water furnished to the radiators on the lower floors

is at a lower temperature, and the amount of heating surface must be correspondingly increased to make up for this loss.

For example, if in the case of Fig. 117 we assume the water to leave at 180 degrees and return at 160, we shall have a drop in tem-
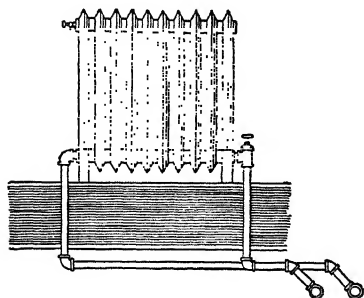

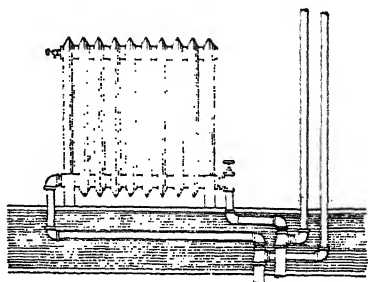Fig. 114.   Radiator Connected with Horizontal Flow and Return Mains Located below Floor

Fig. 115.   Radiator Connected to Vertical Riser and Return Drop

perature of 10 degrees on each floor; that is, the water will enter the radiator on the second floor at 180 degrees and leave it at 170, and will enter the radiator on the first floor at 170 and leave it at 160. The average temperatures will be 175 and 165, respectively.   The
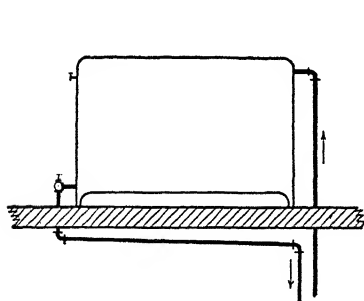

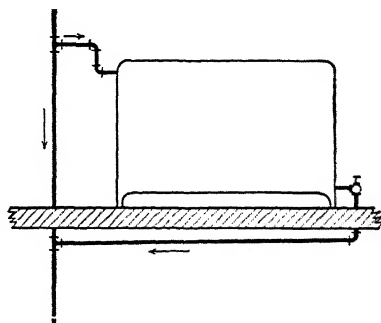Fig. 116.   Upper-Floor Radiator with Supply Connected at Top

Fig. 117.   Radiator Connections, Overhead Distribution System

radiation factor in the first case will be $175 - 70 = 105$; and $105 \times 1.5 = 157$.   In the second case, $165 - 70 = 95$; and $95 \times 1.5 = 142$; so that the radiator on the first floor will have to be larger than that on the second floor in the ratio of 157 to 142, in order to do the same work.

This is approximately an increase of 10 per cent for each story downward to offset the cooling effect; but in practice the supply drops are made of such size that only a part of the water is by-passed
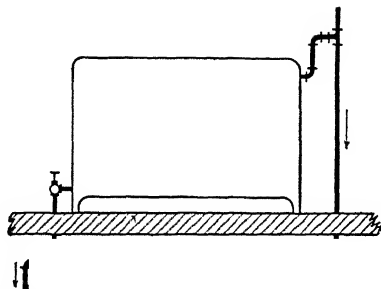

Fig. 118. Another Form of Radiator Connection, Overhead Distribution System

through the radiators. For this reason an increase of 5 per cent for each story downward is probably sufficient in ordinary cases.
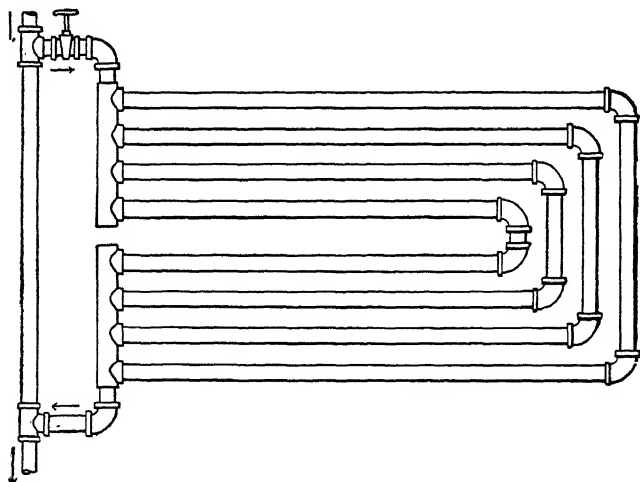

Fig. 119. Pipe Coil for Down Feed System

Where the radiators discharge into a separate return, as in the case of Fig. 108 or in Fig. 111, we may assume the temperature of the water to be the same on all floors, and give the radiators an equal efficiency.

In a dwelling-house of two stories no difference would be made in the size of radiators on the two floors; but in the case of a tall office building, corrections would necessarily have to be made.
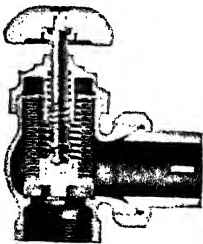

Fig. 120. Packless Hot-Water Valve

Where pipe coils are used, they should be of a form which will tend to produce a flow of water through them. Fig. 119 shows one way of making up and connecting a coil.

Valves and Fittings. Gate-valves should always be used in connection with hot-water piping, although angle-valves may be used at the radiators. There are several designs of radiator valves made

Fig. 121. Equalizing Hot-Water
Radiator Valve

especially for hot-water work. Their chief advantage lies in a device for quick closing, usually a quarter-turn or half-turn being sufficient to open or close the valve. Two different designs are shown in Figs. 120 and 121.

It is customary to place a valve in only one connection, as that is sufficient to stop the flow of water through the radiator; a fitting known as a *union elbow* is often employed in place of the second valve, Fig. 122.

Air=Valves. The ordinary pet-cock air-valve is the most reliable for hot-water radiators. All radiators which are supplied by risers from below should be provided with air-valves placed in the top of the last section at the return end. If they are supplied by drops from an overhead system, the air will be discharged at the expansion tank, and air-valves will not be necessary at the radiators.

Fittings. All fittings, such as elbows, tees, etc., should be of the long-turn pattern. Where pipe sizes are reduced eccentric fit-
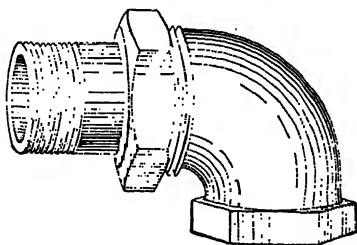

Fig. 122.   Union Elbow

tings should be used. These should be placed so that an air pocket will not be formed at the top of the pipe.

Pipe Sizes. The size of pipe required to supply any given radiator depends upon four conditions; *first*, the size of the radiator; *second*, its elevation above the boiler; *third*, the length of pipe required to connect it with the boiler; and *fourth*, the difference in temperature between the supply and the return.

As it would be a long and rather complicated process to work out the required size of each pipe for a heating system, Tables 61 and 62 have been prepared, covering the usual conditions to be met with in practice.

Table 61 gives the number of square feet of direct radiation which different sizes of mains and branches will supply for varying lengths of run.

It may be used for all horizontal mains.

For vertical risers or drops, Table 62 may be used. This has been computed for the same difference in temperature as in the case of Table 61 (17 degrees), and gives the square feet of surface which different sizes of pipe will supply on the different floors of a building, assuming the height of the stories to be 10 feet. Where

Table 61. Direct Radiating Surface Supplied by Mains of Different
Sizes and Lengths of Run

Size of Pipe	Square Feet of Radiating Surface								
	100 ft. Run	200 ft. Run	300 ft. Run	400 ft. Run	500 ft. Run	600 ft. Run	700 ft. Run	800 ft. Run	1,000 ft. Run
1 inch	30	..	.....	.....	.....	.....	.....	.....	.....
1¼ inch	60	50	.....	.....	.....	.....	.....	.....	.....
1½ inch	100	75	50	.....	.....	.....	.....	.....	.....
2 inch	200	150	125	100	75	.....	.....	.....	.....
2½ inch	350	250	200	175	150	125	.....	.....	.....
3 inch	550	400	300	275	250	225	200	175	150
3½ inch	850	600	450	400	350	325	300	250	225
4 inch	1,200	850	700	600	525	475	450	400	350
5 inch	.....	1,400	1,150	1,000	700	850	775	725	650
6 inch	.....	.....	.....	1,600	1,400	1,300	1,200	1,150	1,000
7 inch	.....	.....	.....	.....	.....	.....	1,706	1,600	1,500

These quantities have been calculated on a basis of 10 feet difference in elevation between
the center of the heater and the radiators, and a difference in temperature of 17 degrees between
the supply and the return.

Table 62. Radiating Surface on Different Floors Supplied by
Pipes of Different Sizes

Size of Riser	Square Feet of Radiating Surface					
	1st Story	2d Story	3d Story	4th Story	5th Story	6th Story
1 inch	30	55	65	75	85	95
1¼ inch	60	90	110	125	140	160
1½ inch	100	140	165	185	210	240
2 inch	200	275	375	425	500	...
2½ inch	350	475	...	...	...	...
3 inch	550	...	...	...	...	...
3½ inch	850	...	...	...	...	...

a single riser is carried to the top of a building to supply the radiators
on the floors below by drop pipes, we must first get what is called
the *average elevation of the system* before taking its size from the
table. This may be illustrated by means of the diagram, Fig. 123.

In *A* we have a riser carried to the third story, and from there a
drop brought down to supply a radiator on the first floor. The
elevation available for producing a flow in the riser is only 10 feet,
the same as though it extended only to the radiator. The water in
the two pipes above the radiator is practically at the same tempera-
ture, and therefore in equilibrium, and has no effect on the flow of the
water in the riser. (Actually there would be some radiation from the
pipes, and the return, above the radiator, would be slightly cooler, but
for purposes of illustration this may be neglected.) If the radiator
were on the second floor the elevation of the system would be 20 feet;
and on the third floor, 30 feet; and so on. The distance which the
pipe is carried above the first radiator which it supplies has but

little effect in producing a flow, especially if covered, as it should be in practice. Having seen that the flow in the main riser depends upon the elevation of the radiators, it is easy to see that the way in which it is distributed on the different floors must be considered. For example, in *B*, Fig. 123, there will be a more rapid flow through
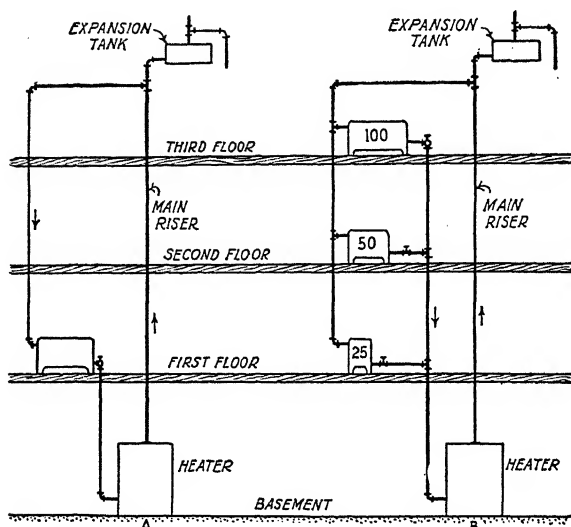

Fig. 123. Diagram to Illustrate Finding of Average Elevation of Heating System

the riser with the radiators as shown, than there would be if they were reversed and the largest one were placed upon the first floor.

We get the average elevation of the system by multiplying the square feet of radiation on each floor by the elevation above the heater, then adding these products together and dividing the same by the total radiation in the whole system. In the case shown in *B*, the average elevation of the system would be

$$\frac{(100\times30)+(50\times20)+(25\times10)}{100+50+25}=24 \text{ feet}$$

and we must proportion the main riser the same as though the whole radiation were on the second floor. Referring to Table 62, we find, for the second story, that a 1½-inch pipe will supply 140 square feet; and a 2-inch pipe, 275 feet. Probably a 1½-inch pipe would be sufficient.

Although the height of stories varies in different buildings, 10 feet will be found sufficiently accurate for ordinary practice.

PRACTICE PROBLEMS

Hot=Water Heating

1. Note Fig. 124*. Calculate the heat losses per room. Assume frame construction, where $\frac{7}{8}$-inch rigid insulation is used as sheathing. Also assume all temperature conditions for your own locality.

(a) Determine the required amount of the best size American Corto radiators for each room in Fig. 124. (See Tables 39 to 42, pages 153 to 156.)
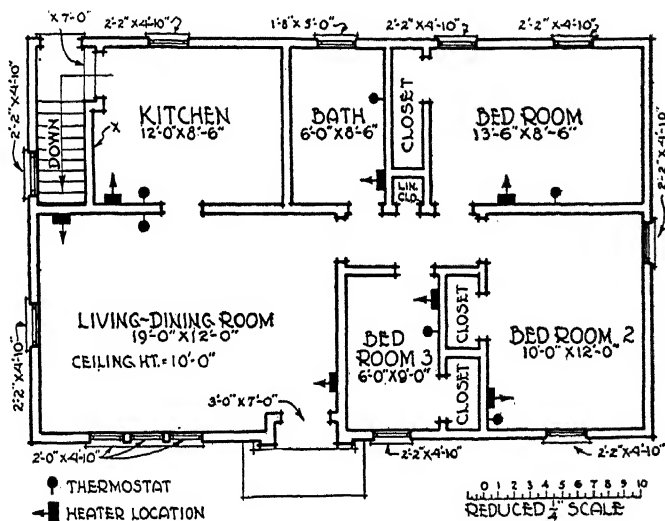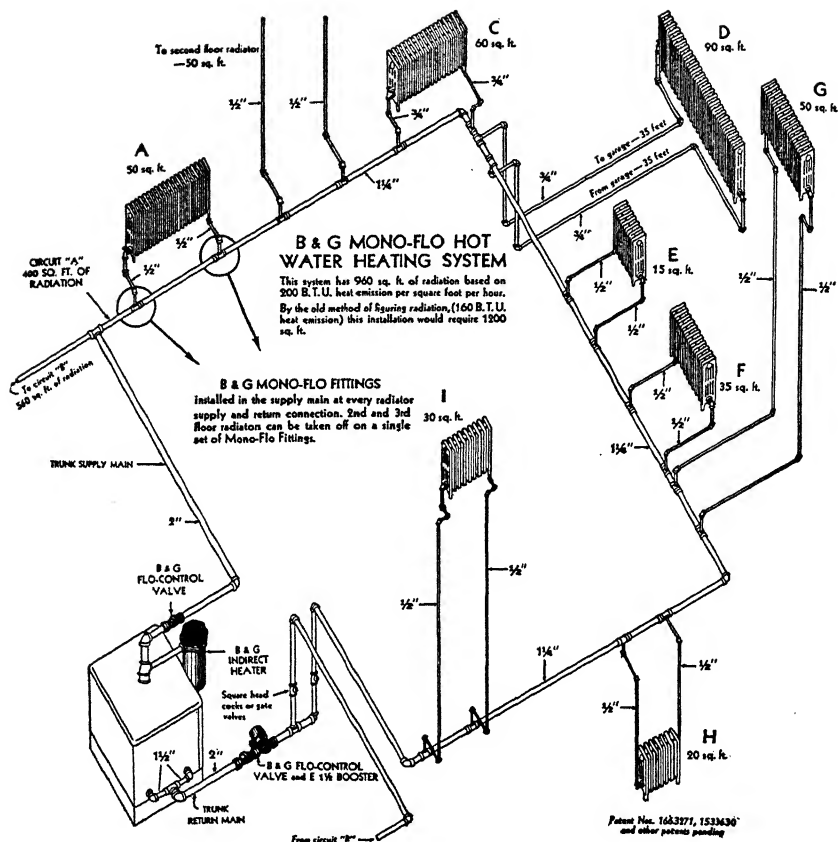

Fig. 124. Small House Floor Plan

(b) Assuming a calorific value of 12,500 B.t.u.'s for soft coal, determine the necessary sizes or numbers of round boilers or sectional boilers—as thought best.

(c) Design a two-pipe up-feed reversed-return, hot-water heating system. Show all pipes and radiators in either plan views or drawings similar to those in the text.

(d) Design the practical size of all pipes shown in the answer to part (c) of this problem.

---

*Disregard thermostats and heaters shown in Fig. 124.

To second floor radiator
—50 sq. ft.

½"   ½"

A
50 sq. ft.

½"

CIRCUIT "A"
400 SQ. FT. OF
RADIATION

½"

1¼"

C
60 sq. ft.

¾"

¾"

D
90 sq. ft.

To garage—35 feet

From garage—35 feet

¾"

¾"

G
50 sq. ft.

½"   ½"

E
15 sq. ft.

½"

½"

F
35 sq. ft.

½"

½"

To circuit "B"
560 sq. ft. of radiation

B & G MONO-FLO HOT WATER HEATING SYSTEM

This system has 960 sq. ft. of radiation based on 200 B.T.U. heat emission per square foot per hour.

By the old method of figuring radiation, (160 B.T.U. heat emission) this installation would require 1200 sq. ft.

B & G MONO-FLO FITTINGS

installed in the supply main at every radiator supply and return connection. 2nd and 3rd floor radiaton can be taken off on a single set of Mono-Flo Fittings.

I
30 sq. ft.

1¼"

TRUNK SUPPLY MAIN

2"

B & G
FLO-CONTROL
VALVE

B & G
INDIRECT
HEATER

Square head cocks or gate valves

1½"   2"

TRUNK
RETURN MAIN

B & G FLO-CONTROL
VALVE and E 1½ BOOSTER

From circuit "B"

½"

½"

1¼"

½"

½"

H
20 sq. ft.

Patent Nos. 1663271, 1533636
and other patents pending

ONE-PIPE HOT-WATER SYSTEM

Courtesy of the Crane Company, Chicago

CHAPTER XI

FORCED HOT=WATER CIRCULATION

The gravity system of hot-water heating is well adapted to buildings of small and medium size. However, there is a limit to which it can be carried economically because of the slow movement of the water, which calls for pipes of excessive size. Forced hot-water circulation overcomes this difficulty by using pumps to force the water through the mains at a comparatively high velocity.

The water may be heated in a boiler in the same manner as for gravity circulation, or exhaust steam may be utilized in a feed-water
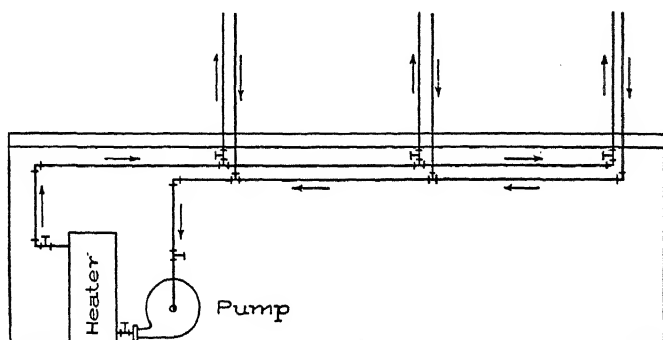
Fig. 125. "Two-Pipe" System for Forced Hot-Water Circulation

heater of large size. Sometimes part of the heat is derived from an economizer placed in the smoke passage from the boilers.

Systems of Piping. The mains for forced circulation are usually run in one of two ways. In the two-pipe system, shown in Fig. 125, the supply and return are carried side by side, the former reducing in size, and the latter increasing as the branches are taken off.

The flow through the risers is produced by the difference in pressure in the supply and return mains; and as this is greatest nearest the pump, it is necessary to place throttle-valves in the risers to prevent short-circuiting and to secure an even distribution through all parts of the system.

Fig. 126 shows the single-pipe or circuit system. This is similar to the one already described for gravity circulation, except that it can

be used on a much larger scale. A single main is carried entirely
around the building in this case, the ends being connected with the
suction and discharge of the pump as shown.

As the pressure or head in the main drops constantly throughout
the circuit, from the discharge of the pump back to the suction, it is
evident that if a supply riser be taken off at any point and the return
be connected into the main a short distance along the line, there will
be a sufficient difference in pressure between the two points to produce
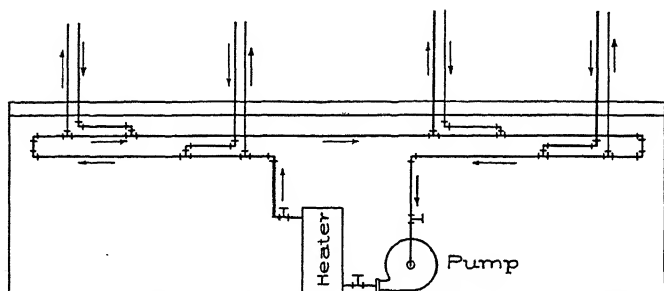
Fig. 126. "Single-Pipe" or "Circuit" System for Forced Hot-Water Circulation

a circulation through the two risers and the connecting radiators.
A distance of 8 or 10 feet between the connections is usually ample
to produce the necessary circulation, and even less is ample if the
supply is taken from the top of the main and the return connected
into the side.

Sizes of Mains and Branches. Because the velocity of flow is in-
dependent of the temperature and elevation when a pump is used,
it is necessary to consider only the volume of water to be moved
and the length of run. The volume is found by the equation

$$Q = \frac{RE}{500\ T}$$

in which Q = gallons of water required per minute; R = square feet
of radiating surface to be supplied; E = radiation factor of radiating
surface in B.t.u. per square foot per hour; and T = drop in tempera-
ture of the water in passing through the heating system.

In systems of this kind, where the circulation is comparatively
rapid, it is customary to assume a drop in temperature of 30 degrees
to 40 degrees between the supply and return.

Having determined the gallons of water to be moved, the required size of main can be found by assuming the velocity of flow, which for pipes from 5 to 8 inches in diameter may be taken at 400 to 500 feet per minute. A velocity as high as 600 feet is sometimes allowed for pipes of large size, while the velocity in those of smaller diameter should be proportionally reduced to 250 or 300 feet for a 3-inch pipe. The next step is to find the pressure or head necessary to force the water through the main at the given velocity. This in general should not exceed 50 or 60 feet, and much better pump efficiencies will be obtained with heads not exceeding 35 or 40 feet.

As the water in a heating system is in a state of equilibrium, the only power necessary to produce a circulation is that required to overcome the friction in the pipes and radiators; and, as the area of the passageways through the latter is usually large in comparison with the former, it is customary to consider only the head necessary to force the water through the mains, taking into consideration the additional friction produced by valves and fittings.

Each long-turn elbow may be taken as adding about 4 feet to the length of pipe; a short-turn fitting, about 9 feet; 6-inch and 4-inch swing check-valves, 50 feet and 25 feet, respectively; and 6-inch and 4-inch globe check-valves, 200 feet and 130 feet, respectively.

Table 63 is prepared especially for determining the size of mains for different conditions.

Example. Suppose that a heating system requires the circulation of 480 gallons of water per minute through a circuit main 600 feet in length. The pipe contains 12 long-turn elbows and 1 swing check-valve. What diameter of main should be used?

Assuming a velocity of 480 feet per minute as a trial velocity, refer to Table 63 and follow along the line corresponding to that velocity and find that a 5-inch pipe will deliver the required volume of water under a head of 4.9 feet for each 100 feet length of run.

The actual length of the main, including the equivalent of the fittings as additional length, is $600 + (12 \times 4) + 50 = 698$ feet. Hence, the total head required is $4.9 \times 6.98 = 34$ feet. As both the assumed velocity and the necessary head come within practicable limits, this is the size of pipe which would probably be used. If it were desired to reduce the power for running the pump, the size of main could be

increased. That is, Table 63 shows that a 6-inch pipe would deliver the same volume of water with a friction head of only about 2 feet per 100 feet in length, or a total head of $2 \times 6.98 = 14$ feet.

The risers in the circuit system usually are made the same size as for gravity work. With double mains, as shown in Fig. 125, they
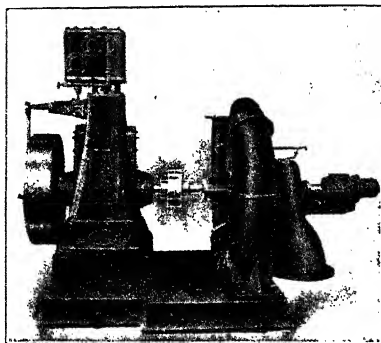

Fig. 127.   Centrifugal Pump Direct-Connected to Engine,
for Forced Hot-Water Circulation

may be somewhat smaller, a reduction of one size for diameters over $1\frac{1}{4}$ inches being common.

The branches connecting the risers with the mains may be proportioned from the combined areas of the risers. When the branches are of considerable size, the diameter may be computed from the available head and volume of water to be moved.

**Table 63.   Capacity in Gallons per Minute Discharged at Velocities of
300 to 540 Feet per Minute—Also Friction Head in Feet,
per 100 Feet Length of Pipe**

Veloc-ity	Diameter of Pipe											
	3-inch		4-inch		5-inch		6-inch		7-inch		8-inch	
	Capac-ity	Fric-tion	Capac-ity	Fric-tion	Capac-ity	Fric-tion	Capac-ity	Fric-tion	Capac-ity	Fric-tion	Capac-ity	Fric-tion
300	110	3.41	195	2.56	306	2.05	440	1.70	600	1.46	783	1.28
480	176	8.16	314	6.12	490	4.9	705	4.08	959	3.49	1,253	3.06
540	198	10.1	352	7.64	550	6.11	794	5.09	1,079	4.36	1,410	3.82

Pumps.   Centrifugal pumps are usually employed in connection with forced hot-water circulation in preference to pumps of the piston or plunger type. They are simple in construction, having no valves; they produce a continuous flow of water; and, for the low

heads against which they are operated, they have a good efficiency.
A pump of this type, with a direct-connected engine, is shown in
Fig. 127.

Under ordinary conditions the efficiency of a centrifugal pump
falls off considerably for heads above 30 or 35 feet; but special high-
speed pumps are constructed which work with a good efficiency
against 500 feet or more.

Under favorable conditions an efficiency of 60 to 70% is often
obtained, but for hot-water circulation it is more common to assume
an efficiency of about 50% for the average case.

The horsepower required for driving a pump is given by the
following formula:

$$\text{hp.} = \frac{H \times V \times 8.3}{33,000 \times E} \tag{30}$$

in which H = friction head in feet; V = gallons of water delivered per
minute; and E = efficiency of pump.

Centrifugal pumps are made in many sizes and with varying
proportions, to meet the different requirements of capacity and head.

Heaters. If the water is heated in a boiler, any good form may
be used, the same as for gravity work. In case tubular boilers are
used, the entire shell may be filled with tubes as no steam space is
required.

To prevent the water from passing in a direct line from the
inlet to the outlet, a series of baffle-plates should be used to bring it
in contact with all parts of the heating surface.

When steam is used for heating the water, it is customary to
employ a closed feed-water heater with the steam on the inside of the
tubes and the water on the outside.

Any good form of heater can be used for this purpose by provid-
ing it with steam connections of sufficient size. In the ordinary form
of heater, the feed-water flows through the tubes, and the connec-
tions are therefore small, making it necessary to substitute special
nozzles of large size when used in the manner here described.

When computing the required amount of heating surface in the
tubes of a heater, it is customary to assume a transmission factor
of about 200 B.t.u. per square foot of surface per hour per degree
difference in temperature between the water and steam.

It is usual to circulate the water at a somewhat higher tempera-
ture in systems of this kind, and a maximum initial temperature of
200 degrees, with a drop of 40 degrees in the heating system, may be
used in computing the size of heater. If exhaust steam is used at
atmospheric pressure, there will be a difference of $212-180=32$
degrees, between the average temperature of the water and the steam,
giving a factor of $200\times32=6,400$ B.t.u. per square foot of heating
surface.

From this it is evident that $6,400\div170=38$ square feet of direct
radiating surface, or $6,400\div400=16$ square feet of indirect, may
be supplied from each square foot of tube surface in the heater.

Example. A building having 6,000 square feet of direct, and 2,000 square
feet of indirect radiation, is to be warmed by hot water under forced circulation.
Steam at atmospheric pressure is to be used for heating the water. How many
square feet of heating surface should the heater contain?

$6,000\div38=158$; and $2,000\div16=125$; therefore $158+125=283$
square feet, the area of heating surface called for.

When the exhaust steam is not sufficient for the requirements,
an auxiliary live steam heater is used in connection with it.

Pipe Expansion. The proper provision for the expansion and
contraction of piping must be made in all cases where water or
steam is to be used at high temperatures, and is usually accomplished
by long sweep bends or expansion joints. Certain joints, usually
where branches are taken off, are securely anchored to the building
structure, and the movement between these points taken up by
the expansion members, such as bends or joints. Where steam
mains run great distances there is generally a large "U" shape
bend made in them at specified places. These bends take up any
amount of expansion without causing any movement of the main
line.

Expansion of piping is ordinarily based on the theoretical
elongation of the measured length of the line for the difference in
temperature between the air at the time the pipe was fitted and the
final temperature when filled with steam or hot water.

CHAPTER XII

AUTOMATIC CONTROLS

The process of heating a building consists of supplying sufficient heat to the building to compensate for its constant loss of heat and to maintain certain uniform temperatures during the hours of occupancy.

Similarly, the process of cooling a building consists of supplying sufficient cooling to the building to compensate for its constant heat gain from the outdoors, occupants, sun, lights, etc., and to maintain certain uniform temperatures during the hours of occupancy.

The heating or cooling system must be designed to heat or cool the building to its desired temperature regardless of the outdoor temperature. Inasmuch as extremely cold or hot weather represents less than 5% of the heating or cooling season, it is evident that the heating or cooling system need not be operated at full capacity 100% of the time. During the remaining 95 or greater per cent of the time, therefore, the release of heat or cooling must be controlled, to prevent overheating or overcooling and the resultant waste.

In addition to the control of temperature, the control of relative humidity and of air distribution are important for a satisfactorily conditioned space. An automatic control system directs the operation of each of the various portions of the system. An air-conditioning system without all the necessary controls cannot be expected to operate properly.

Control Systems for the Heating Cycle. Control systems used for the heating cycle, or winter operation, as it is sometimes called, vary with the type and size of the building, occupancy of the building to be heated, and also with the heating system, humidity supplying equipment, and ventilating means available for control. In the following the general phases will be discussed.

Single Thermostat Control. Probably the most widely used form of control is that regulated entirely from a single room thermostat. The wide use of this particular means of control is due to the

fact that it is the form best adapted to residences and small buildings, which far outnumber the larger structures. In larger buildings, this form of control has definite shortcomings. In the small buildings and average size residences it is possible to select a location and install a thermostat which, in controlling from the surrounding air temperature, will hold the temperature of the entire building within satisfactory limits. In such buildings, provided the delivery of heat is well balanced and distribution is good, conditions throughout the whole structure will be represented by the conditions at the thermostat location. The thermostat reacts to and controls from the temperatures to which it is subjected and the positions selected for the thermostat must be representative of general conditions throughout the structure. If certain areas of a structure are not properly balanced as regards heating capacity and distribution, the control dictated by the thermostat will not produce satisfactory results in these unbalanced areas. As an example, if in a two-story home there is a lack of heating capacity on the second floor, a thermostat located in the living quarters on the first floor cannot be expected to compensate for such lack, and these rooms will normally assume a lower temperature level than that dictated by the thermostat. On the other hand, rooms in which more than sufficient heating capacity is provided will overheat. The corrections for these conditions will be the proper balancing of heat distribution or the addition of some separate heat supplying means or automatic control function which will correct the unbalanced condition. As the building becomes larger, the dependence on proper heat balance becomes greater if a single thermostat control system is to be used in regulation of the entire structure. Under the heading of "Zone Control," further consideration is given the handling of larger buildings.

Individual Room Control. The most accurate and flexible form of control for any structure is in the regulation of each room by control equipment reacting to conditions in that room only. Such control consists of a thermostat handling an individual radiator valve, a unit ventilator, unit heater, a radiator equipped with a self-contained radiator valve, or similar heating source supplying heat only to the room in which it is installed. Thus, the controlling device will regulate the amount of heat supplied to the individual room, and

temperature levels of any room may be maintained without reference to temperature levels in other rooms of the same structure. This form of control, due to the number of control devices required over the entire building, normally is the most expensive type of control system. However, where maximum flexibility and the most accurate control is desired, individual room control furnishes the desired results.

Zone Control. As the buildings increase in size, it becomes increasingly difficult to provide proper regulation from single thermostat control, and the expense of individual room control is often not justified. An intermediate form of control system is available and is described as zone control. In this system a building is divided into areas or zones in which the general requirements and the general conditions throughout the areas are relatively constant. Each zone is provided with control equipment which functions to regulate the conditions in that particular zone. As in the case of individual room control, each zone may be regulated to its own needs. Thus zone control, while providing a less accurate form of control than individual room control, does provide a greater flexibility than single thermostat control and at the same time the expense is held within reasonable limits.

There are several considerations involved in the proper division of a building into zones. Weather conditions have an important bearing on heat distribution, so one of the primary considerations in zoning must be that of exposure. In the United States, and particularly in the northern portion, the prevailing winds are from a northerly direction. Thus, during the heating cycle the northern faces of the buildings are subjected to the greatest effect of wind. Building faces subjected to wind will have a greater heat loss than exposures not subject to wind. The rooms or zones on the northern exposure of the building will require a greater amount of heat than those on the less exposed faces. The wind velocity enters into this consideration because the heat loss is directly proportional to the wind velocity.

Outside temperature determines the temperature gradient between the interior and the exterior of a building, and the heat loss is directly proportional to this gradient. Thus, the lower the outside temperature the greater the heat loss and the greater the

amount of heat which must be supplied in the building to maintain any given temperature.

Within the last few years the effect of solar radiation has come into prominence in consideration of the heating problem, as it has been recognized that portions of the building exposed to the effects of the sun have a lower heat loss than those which are constantly in the shade. Here again it follows that if the sun effect reduces the heat loss over a portion of the building, this portion of the building requires less heat input to maintain any given temperature level.

It is not possible to segregate these four factors—outdoor temperature, wind direction, wind velocity, and solar radiation—as they exert a combined effect as well as a variable effect on every building structure. Therefore, there is a constantly changing heat loss from each building which must be measured and reduced to terms of regulation of the heat supply if the temperature levels throughout the building are to be maintained within set limits and with minimum operating cost.

In terms of zoning, it does follow that the conditions of greatest heat loss generally will be found on the north face of the building, due to the fact that here will be found the prevailing winds with varying velocities with a minimum of sun effect. The east and west faces of the building, while subject to some wind, are also the beneficiaries of some sun effect and the total heat loss is, therefore, generally less than that of the north face. Similarly, the south face of the building subjected to less wind effect and most sun effect normally has the smallest heat loss and, accordingly, requires the smallest heat input. Thus, the first consideration in zoning the building is to make a division based on the four directional exposures, and in applying temperature control to design the heat distributing means with a view to supplying heat to these zones based on the needs of their particular exposures.

Other considerations in the zoning of any building are based on the occupancy of different parts of the building. In a large residence it may be less important in terms of comfort of the occupants or less important from an operating cost standpoint to zone the building for exposure than it is to zone the building for occupancy. As an example, in a large residence it may be more desirable

to divide the building into zones consisting of living quarters, sleeping quarters, nursery, servants' quarters, guest quarters, and garage than to zone the building on a basis of weather exposure. With the above mentioned divisions the primary consideration is the maintenance of suitable temperature levels, which may vary considerably. Thus, the living quarters may be maintained at temperature levels of 70°F. to 72°F., while the nursery may require temperature levels of 74°F. to 76°F.; the garage, temperature levels of 50°F. to 55°F., etc. Similarly, in an industrial building such a division may be made between office quarters, factory quarters, and warehouse areas, with the office maintained at approximately 72°F., the factory at approximately 65°F., depending on the type of work going on, and the warehouse at some lower temperature depending upon what is stored therein. Where the temperature levels vary greatly and it is possible to hold sizable areas at reduced temperatures, the cost of heating may be lower when zoning on this basis than when zoning on the basis of exposure.

Zoning need not be restricted to either one or the other of these general considerations, because it is often possible to zone a building with both considerations in mind, the primary requisite being that the distributing means be sufficiently divisible that maximum flexibility permissible with the control devices be attained.

In either of the above types of zoning or in combination zone layouts, the time factor may also be included. Thus, in a residence, the temperature levels may be determined and regulated on the basis of comfort conditions during the hours that the occupants are active and utilizing these quarters. At other hours, normally the night hours, when some or all of these zones are not utilized, it is possible to effect substantial savings in fuel expense by automatically changing temperature levels to a lower degree. Automatic controls such as clock thermostats and time switches permit such changes in temperature level to be adjusted to any hourly cycles desired, and flexibility of modern temperature control systems is such that within the limits of the zoning, predetermined automatic adjustments based on time may be made.

Where relatively large areas are included in zone control, it becomes increasingly difficult to locate a single thermostat which will measure and regulate the entire zone area properly. This is

due to the difficulty of selecting a point that represents average conditions which will permit the thermostat to be installed where it will not be subject to upsetting influences such as open windows or tampering by unauthorized persons. One form of system to overcome the difficulty of regulating a large zone from one location makes use of two or more thermostats so interconnected that the temperature measured at their several locations is averaged. This average or resultant temperature level then becomes the control temperature to be maintained.

Another attack on these difficulties has led to the development of several types of systems which avoid the necessity of depending on a key location thermostat. There are various forms of these systems but primarily they are designed to regulate the heating system from outdoor conditions. In several systems the practice followed is that of measuring outdoor temperature and adjusting the heat input in reverse ratio to the outdoor temperature. Thus, as the outdoor temperature drops, a proportionately greater amount of heat is distributed to the zone. Several such systems obtain their results by varying the control from outdoor temperature only, or a combination of outdoor temperature and solar radiation. Still other zone control systems make use of devices which react to the combined effects of outdoor temperature, wind direction, wind velocity, and solar radiation in determining the heat requirements of the zone which they control. Zone control systems provide satisfactory temperature control and effect substantial reductions in operating costs through decreased consumption of fuel.

Controls for the Cooling Cycle. In general, the questions of control in the cooling cycle follow closely those of the heating cycle. Thus the advantages or disadvantages of single thermostat control, individual room control, or zone control must receive similar consideration except that cooling necessitates the reduction rather than the addition of heat and humidity.

Dehumidification Controls. During the cooling cycle one of the major considerations is that of dehumidification of the air when the relative humidity increases beyond the limits of comfort. Except in the case of chemical absorption of moisture from the air, all other forms of dehumidification depend primarily on reducing the air temperature to a predetermined degree at which the absolute humidity

is reduced, due to condensation, to some predetermined and accept-able value. After such reduction of the absolute humidity, the air sometimes must be warmed again and delivered to the conditioned area sufficiently dried out to assure a satisfactory and comfortable relative humidity. Because the air has been subjected to the de-humidification process, it sometimes is so lowered in temperature that it cannot be delivered directly into the conditioned area or zone. In this case a proper balance is established by infusing warmer air.

From the control standpoint, the need for dehumidification and the actuation of dehumidifying devices may be measured and con-trolled by reverse acting humidity controls, these units starting and stopping compressors, air washers, etc. Or these controllers may operate damper motors which dictate the distribution of air through and around cooling coils.

All=Year Control. With the increasing emphasis on air condi-tioning as a year-around program, there will be an increase in the design and installation of systems which properly treat the air at all seasons. While the modernization type of job with an already existing heating system, to which the cooling cycle equipment will be added, may be operated in two phases, one to cover the heating cycle and the other to cover the cooling cycle, the installation of automatic control will make it desirable that these two phases be interlocked so that there will be a minimum amount, if not complete elimination, of the need for manual attention in arranging the system to function on either cycle. There are, during the days of late spring and early fall, periods within which the complete system may be required to change from the heating to the cooling cycle and back again during a single day. For instance, during these seasons it is often necessary that heat be supplied during the night or in the early morning or evening hours, but, during the midday, temperatures and humidity conditions are such that cooling cycle operation is desirable. Where the control system or the design of the primary equipment is such that it is a burdensome task to change the system in accordance with these needs, the satisfaction to the user cannot be as complete as in the case of a system which can and will make changes as frequently as re-quired and in a completely automatic manner.

From the standpoint of the automatic controls, such automatic

change-over is entirely practical and available provided the air conditioning equipment has been selected and arranged with this possibility in mind.

The Residence. Considering the residence, the control installation may vary from the simple temperature regulation of a coal-fired heating plant to the completely automatic all-year air-conditioning installation. Where temperature control of the heating cycle is the only problem, the selection of control equipment is based on the type of heating plant, the temperature requirements and the heat source. The type of heating plant will dictate the form of limit control to be used, and the heat source, i.e., oil burner, gas burner, stoker, will legislate the selection as to the automatic sequence and safety controls required.

The temperature requirements to be met will determine whether a plain pattern thermostat, maintaining temperature levels continuously at the thermostat setting, or a clock pattern thermostat, which lowers the temperature level during the night, is to be used. Another question which has a bearing on the selection is the relative location of the thermostat to the controlled devices, as it usually is possible to lower the installation cost by selecting control equipment using low voltage wiring if such wiring runs are very long. It is true that the practice of manufacturers to provide control equipment as a part of their product will relieve the engineer or architect of some of the above decisions, but in the interests of insuring both the results and the safety of the system, the control in terms of results should be specified.

As the size of the house increases, the regulation must be considered in terms of flexibility of thermostat control, hence the selection with reference to outcome or effectiveness must include the consideration of individual room or zone control.

A review of the preceding paragraphs on the subject of individual room control and zone control will point out the particular advantages of each form of control and the factors which must be taken into account in making a selection.

When the residence includes provision for other than simple heating plant functions, the control system must be broadened accordingly. With the addition of humidity supplying equipment, the control specification should recognize the desirability of regulating

the relative humidity within safe limits, and as fan distribution systems are normally involved in this type of installation, the humidity control should be interlocked so the humidifier is not operated unless the fan is running. Where the maximum refinement in humidity control is desirable, the compensated type of control which lowers the relative humidity point in cold weather can be added. Where the residence installation involves fan systems, provision for the auto-
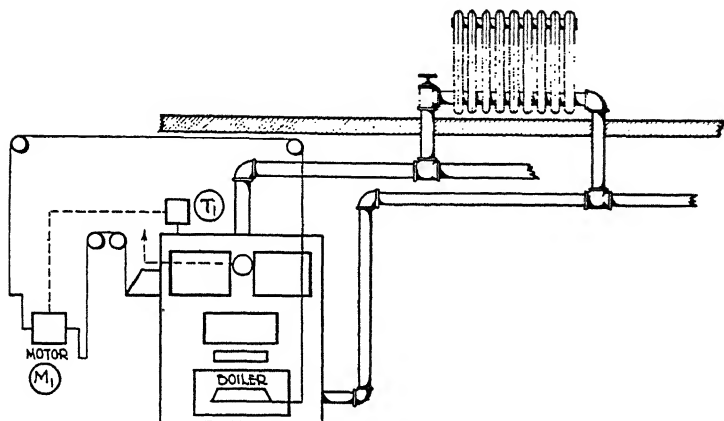

Fig. 128. Controls for Coal-Fired Steam or Hot-Water Heating Plant

matic control of the fresh air and recirculated air dampers, heating of cooling coils and air washers must be considered with respect to type of heating and the general characteristics of the distribution system. Such systems used in connection with cooling means in the summer will require control of the refrigeration means and decisions as to the desirability of adjusting the indoor temperature manually or raising it automatically as outdoor temperature rises. Again it may be desirable to include complete effective temperature control. In localities where high relative humidities produce uncomfortable conditions during the cooling season and where the cooling means is capable of dehumidification, automatic control selecting periods during which the dehumidification is to be put into operation will be desirable. Where separation of the sensible cooling and dehumidification can be effected in the design of the primary equipment, the cost of operation may be lowered, as the removal of the existing moisture is the more costly of the two processes.

Steam and Hot=Water Heating Systems. The control systems for either steam or hot-water heating plants are similar in so many ways that they may be considered as one group. The control system of each of these differs only in that the controllers and measuring devices must be suitable for steam or for hot water.

In many respects, the control of a forced air heating system with a steam or hot water boiler as the heat-producing source, must provide the same synchro-
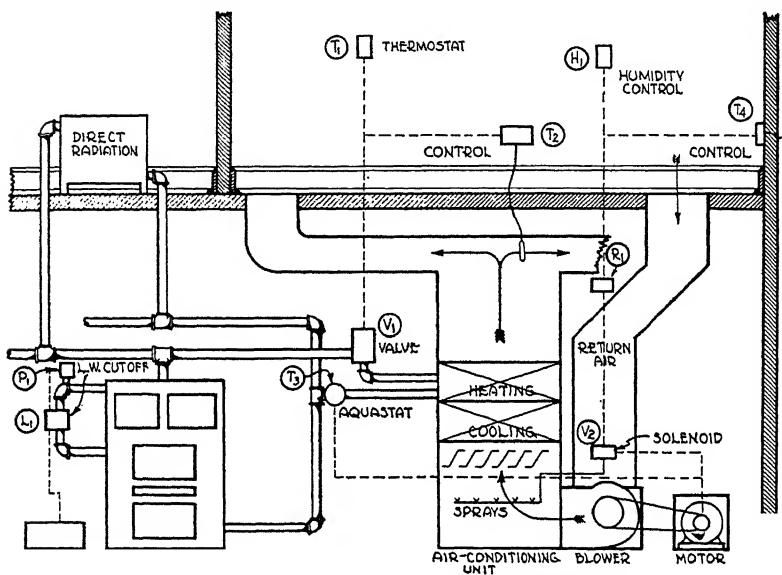

Fig. 129. Controls for an Automatically Fired Steam Heating Plant Using both Radiation and Forced Air

nized functions that are necessary with a system securing its heat from a warm air furnace. Therefore, many of the systems of control may be combined with those systems that are peculiar to boiler operated plants to produce a complete system of control for the more elaborate type of installation.

Fig. 128 illustrates a system of control for steam or hot-water heating plants of simple design. The operation of the control system is as follows· Limit control T_1, operates motor M_1, which positions the check and draft dampers in accordance with the temperature or pressure changes. Definite conditions are maintained within the approximate differential of the limit control.

Fig. 129 illustrates a system of control for an automatically fired steam heating plant, using both direct radiation and forced warm air for heating. This is known as a combination or split system. Temperature control is provided for the portion of the building heated by forced warm air, and thermostatic control for the portion of the building heated by direct radiation should also be provided. The humidity control is automatically compensated by an outside temperature compensator.

*See Chapter VI for other control systems.

The operation of the control system is as follows: Pressure controller P_1 will maintain a definite steam pressure in the boiler. Low water cutoff L_1 will automatically stop the oil burner if the water level in the boiler falls below the safe operating level. Through the operation of these controls, steam will always be available for the direct radiation, and for the heat transfer coils in the air-conditioning unit.
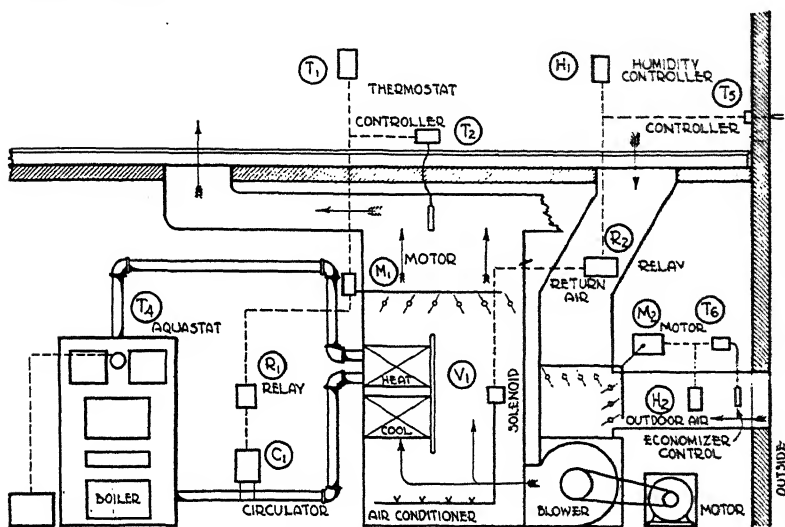

Fig. 130. Controls for an Automatically Fired Hot-Water Plant Using Forced Warm Air for Heating

Thermostat T_1 will measure the temperature of the room, and will operate the steam valve V_1, to admit steam to the heat exchanger in the air-conditioning unit. The control will permit enough steam to enter the heat exchanger so that the air passing through the heat exchanger will be heated sufficiently to offset the heat losses of the room. Temperature controller T_2, located in the plenum chamber, or main discharge duct, will act as a low limit to prevent delivery of air at a temperature too low for satisfactory conditions.

Temperature controller T_3 will permit the blower to operate when the hot condensate from the heat exchanger in the air-conditioning unit has started to flow and has raised the temperature of the water in the discharge line.

Humidity controller H_1 will measure the humidity conditions in the room and will operate relay R_1. Its control point will be varied by outdoor compensator T_4 as the outside weather conditions change. Relay R_1 controls the action of solenoid valve V_2, which will admit water to the sprays in the humidifying equipment when humidity controller H_1 indicates that additional humidity is required. The humidity controls should be so arranged electrically that the water valve V_2 can be operated only when the blower is in operation.

Other systems of control can be added to this system to make it still more complete. A similar system for hot water can be built up from suitable sub-systems of control.

Fig. 130 illustrates a system of control for an automatically fired hot-water heating plant using a forced warm air heating system to supply all the heat necessary for the building. There is no direct radiation. Air temperatures at the registers are varied to meet the heat losses from the building, while the amount of air delivered from the registers is constant. Humidity control is compensated with the outside weather conditions and an economizer control provides for the use of outside air during the heating cycle whenever its use will decrease the load on the heating plant.

The operation of the control system is as follows: Temperature controller T_4 will operate over its differential and maintain a definite boiler water temperature continually. Circulator C_1 will circulate the hot boiler water through the heat exchanger in the air-conditioning unit. There is no flow valve in the hot water line leading from the boiler, and, consequently, there will be a limited amount of gravity circulation when the circulator C_1 is not in operation. This circulation will be sufficient to keep the heat exchanger warm continually. An optional arrangement would be to place a flow valve in the hot water line from the boiler, in which case circulation would cease as soon as the circulator C_1 stopped.

Room thermostat T_1 will measure the room temperature and operate motor M_1 to position the mixing dampers between the tempered air plenum chamber and the recirculated and warm air plenum chambers, so that the delivered air temperature will be suitable to offset the structural heat losses. Temperature controller T_2 located in the tempered air plenum chamber will act as a low limit control to prevent the circulation of air at too low a temperature.

When motor M_1 is in the position where the dampers to the warm air plenum chamber are entirely closed, the auxiliary switch on motor M_1 will stop the operation of circulator C_1 through the action of relay R_1. There will be some gravity circulation of the hot boiler water through the heat exchanger in the air-conditioning unit.

Humidity controller H_1 will measure and regulate the relative humidity of the room, but its control point will be varied to suit the outside weather temperature through the outdoor compensator T_5. Humidity controller H_1 will operate relay R_2 which will open solenoid water valve V_1 and permit the flow of water to the humidifying equipment. Provision should be made so that the solenoid water valve V_1 cannot open if the blower is not in operation.

Economizer controller T_6 will measure the outside temperature and will position motor M_2 and the mixing dampers in the return air ducts, thereby admitting outside air to be mixed with the return air from the building in varying amounts according to the outside temperature. This arrangement permits the introduction of outside air when it is warm enough to reduce the load on the heating system.

The blower will operate continuously, subject only to the control of a manually operated switch. An alternate control is suggested that will stop the blower when the room thermostat calls for no heat and has placed the damper to the warm air plenum chamber in the fully closed position.

A typical method of chilling water by ice is illustrated in Chapter XVII (Fig. 230). The control for this system is as follows: Three-way valve 1, controlled by either a room or duct type thermostat, permits either chilled water or recirculated water to be circulated through the cooling coils to maintain the

desired temperature. It operates in such a manner that on a call for heat the valve admits all recirculated water and on a call for cooling it admits all chilled water.

Forced Warm Air Heating Systems. Fig. 131 illustrates a system of control for a coal-fired, forced warm air heating plant in which the room thermostat controls the operation of the heat generating equipment and the blower. A low limit control will prevent delivery of low temperature and a high limit
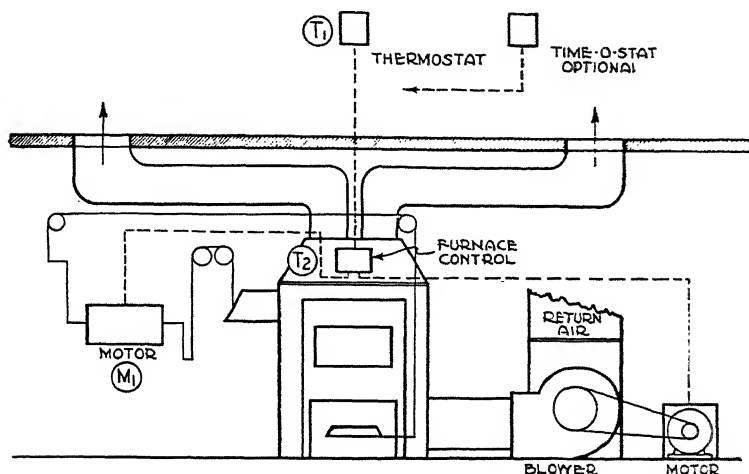

Fig. 131. *Control System for a Coal-Fired Forced Warm Air Heating Plant*

control will take command of the fire in case the bonnet temperatures reach an excessive point. Temperature safety over-run protection is provided to protect the furnace when the blower is not operating. The heat actuated thermostat is used with this system of control.

The operation of the control system is as follows: A fall in room temperature will cause thermostat T_1 to accelerate the fire and attempt to start the blower through the action of program control T_2. Program control T_2 will operate motor M_1 to reposition the draft and check dampers to accelerate the fire. Upon a definite rise in bonnet temperature the low limit control of program control T_2 will permit the operation of the blower and delivery of heat to the room. Upon a rise in room temperature motor M_1 will reposition the dampers to reduce the fire, and the blower will stop, through the action of temperature controller T_2, thereby preventing further delivery of heat.

Program controller T_2 is equipped with a temperature over-run safety, so that if for any reason the operation of room thermostat T_1 has failed to reduce this fire after the blower has stopped, and an excessive bonnet temperature is developed, the blower will start and will dissipate the excess heat into the rooms. This will prevent damage to the heating plant.

This system of control will automatically operate the draft damper so as to reduce the fire in case of line voltage failure. A power failure will cause the blower to stop, and the limited gravity circulation of air through the heating

plant will not prevent hazardous temperatures from developing unless the fire is
immediately checked.

The use of the heat actuated thermostat will cause frequent cycling of the
controls associated with it. This will provide a definite "ceiling" for the fire, which
will vary with the severity of the outside weather conditions.   The frequent
blower operations will tend to prevent stratification and maintain room tempera-
tures within a narrow differential.   This control system provides all the necessary
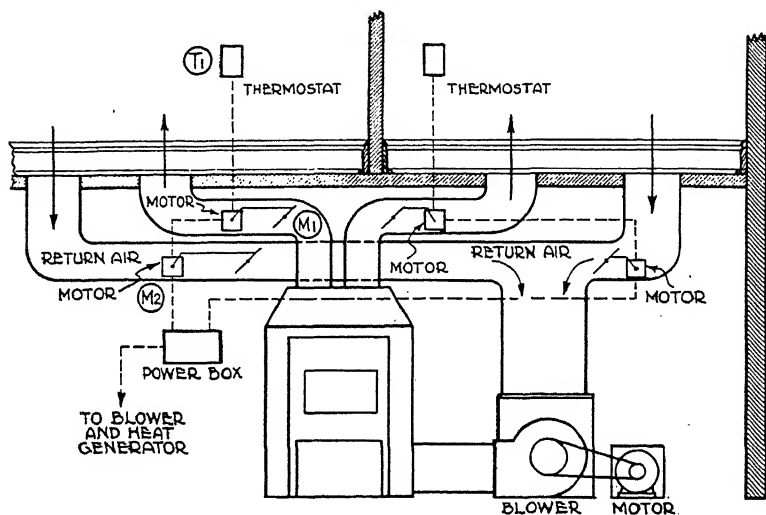

Fig. 132. Zone Control System

functions for the satisfactory control of the average residential forced warm air
heating plant.

Fig. 132 illustrates a zone control system in which there are motor operated
dampers in both the warm air and return air ducts for each zone. The purpose of
this control is to limit the amount of pressure that can be built up in the plenum
chamber as the individual warm air duct begins to close off in response to the
room thermostat.

The operation of this control system is as follows: On a rise in room tempera-
ture, the action of the room thermostat T_1 causes the damper motor M_1 to close
the damper in the warm air duct to the zone. The operation of motor M_1 also
operates the auxiliary switch on the motor, and through its closing causes motor
M_2 to operate and close a damper in the associated return air duct. Control
wiring from the auxiliary switch on motor M_2 on the return air duct is brought to
the power box and from there to a program control.

It is conceivable that on a four zone job, three of the largest ducts may be
in the closed position, leaving the duct damper of a comparatively small zone in
the open position. Under this condition the entire air volume output of the fan
and the entire thermal output of the heater would be concentrated on this one
zone. The fact that as the flow of air increases through a duct the resistance
increases approximately as the square of the increase in velocity provides a

limiting factor as to the actual quantity of air that the blower will force into this single zone. On a properly engineered and installed job, with all zone ducts of about equal length and of equal resistance with no particularly short ducts, the increased air delivery through the registers of this one zone is not likely to be objectionable. On the other hand, if the damper in the open position should be one in a fairly large duct, or in a rather short duct, it is quite probable that air would be discharged at excessive velocity from the room registers.

Placing a damper in the return air and closing it when the warm air damper is closed, somewhat restricts the delivery of the fan by reducing the amount of
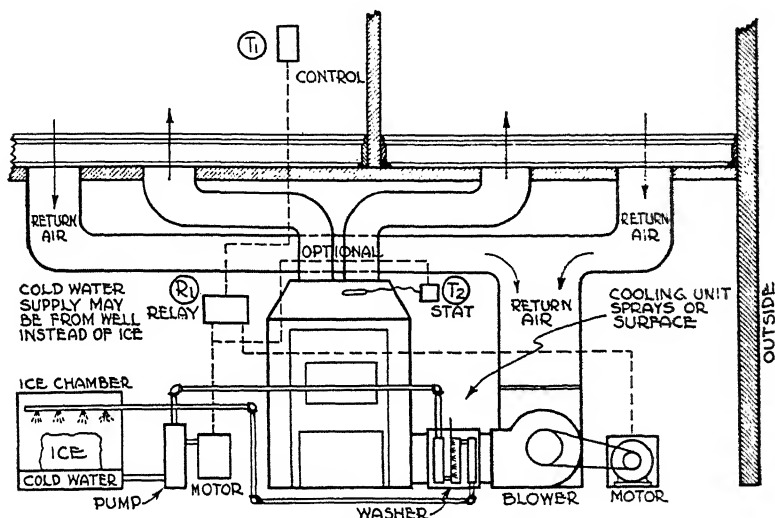

Fig. 133. Control for Cooling System Equipment in an All-Year Furnace

return air it can take in. This will reduce the air volume output of the blower and will maintain normal velocities in the remaining ducts. Placing a damper in the return air also confines the circulation of air through the system to those rooms not yet up to temperature. A system without a damper in the return air will attempt to draw air equally from all rooms of the structure, subject, of course, to the limiting factors of the ability of the air to move freely and unrestrictedly throughout the house.

All-Year Furnaces. Fig. 133 illustrates a system of control for the cooling equipment of an all-year residential conditioning system in which cold water from a well or an ice chamber is passed through a heat exchanger located in the return duct system.

The operation of the system is as follows: Thermostat T_1 measures the relative humidity and the temperature in the room. When there has been an increase in temperature or a rise in relative humidity over the control setting of thermostat T_1, relay R_1 will start the circulating pump and the blower. Temperature controller T_2 will act as a limit control to stop the circulating ice water pump if the discharged air in the plenum chamber becomes so cold that unsatis-

factory conditions will result. Under these conditions, however, the blower will not stop except through the action of room thermostat T_1.

If the cooling equipment uses water from a deep well instead of circulated water from an ice chamber, relay R_1 will start the deep well pump instead of the circulating pump and the water would be passed to the drain instead of being recirculated.

It is not always necessary to use a limit control T_2 in the discharge air, but if the cooling equipment has sufficient capacity to reduce the air temperature to a
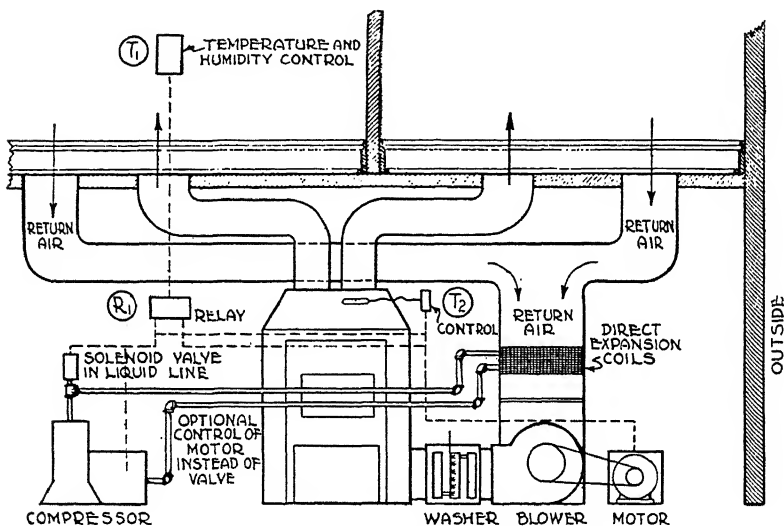

Fig. 134. Control System for All-Year System Using Mechanical Refrigeration

low enough point to produce unsatisfactory conditions in the room, the use of a limit control is preferred. When the cooling equipment uses water sprays instead of surface type coils, it is advisable to use a combination controller that will stop further action of the sprays if the relative humidity of the room rises to an unsatisfactory point when the sprays are operating.

Fig. 134 illustrates a system of control for an all-year residential air-conditioning system in which a mechanical refrigeration system is used in conjunction with a direct expansion coil located in the return air duct system.

The operation of this control system is the same as the control system illustrated in Fig. 133, except that instead of controlling the motor of a circulating pump, a solenoid valve in the liquid refrigerant line of the compressor is controlled.

The compressor may be controlled by direct control of the compressor motor if preferred. Some systems of mechanical refrigeration require the maintenance of a back pressure at the compressor within definite limits, and operate the compressor motor by a refrigeration pressure controller. In such cases a refrigerant solenoid valve which is operated by the control system is placed in the liquid line between the compressor and the direct expansion coils.

CHAPTER XIII

VACUUM SYSTEMS

In the systems of steam heating which have just been described, the pressure carried has always been above that of the atmosphere, and the action of gravity has been depended upon to carry the water of condensation back to the boiler or receiver; the air in the radiators has been forced out through air-valves by the pressure of steam back of it. Methods will now be taken up in which the pressure in the heating system is less than the atmosphere, and where the circulation through the radiators is produced by suction rather than by pressure. Systems of this kind have several advantages over the ordinary methods of circulation under pressure. *First,* no back pressure is produced at the engines when used in connection with exhaust steam; but rather there will be a reduction of pressure due to the partial vacuum existing in the radiators; *second,* there is a complete removal of air from the coils and radiators, so that all portions are steam-filled and available for heating purposes; *third,* there is complete drainage through the returns, especially those having long horizontal runs; and there is absence of water-hammer; and *fourth,* smaller return pipes may be used. The two older systems of this kind in common use are known as the Webster and Paul systems.

Webster System. The Webster system consists primarily of an automatic outlet-valve on each coil and radiator, connected with some form of suction apparatus, such as a pump or ejector. One type of valve used is shown in section in Fig. 135, which replaces the usual hand-valve at the return end of the radiator. It is similar in construction to some of the air-valves already described, consisting of a bellows or sylphon, which is filled with a volatile liquid; in the presence of the steam the liquid partially vaporizes, thus expanding the bellows so that it presses against the valve opening and closes it. When water or air fills the valve, the bellows contracts and allows it to be sucked out as shown by the arrows.

Fig. 136 shows a thermostatic valve or trap which operates on the same principle as the one shown in Fig. 135, but which is designed
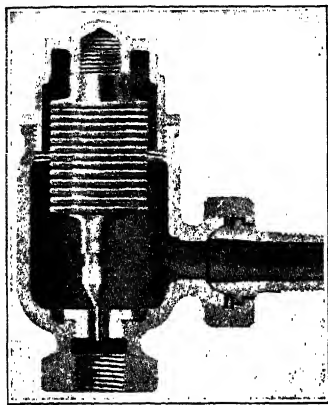

Fig. 135. Webster Air and Water Outlet-Valve or Trap for Radiator
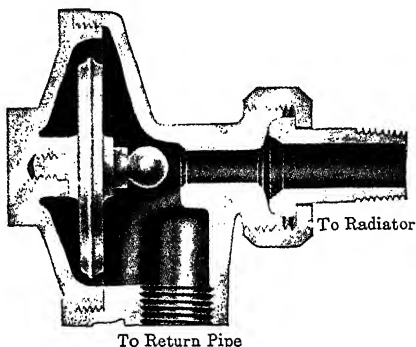*Courtesy, Warren Webster and Company, Camden, New Jersey*

To Radiator

To Return Pipe

Fig. 136. Illinois Thermostatic Radiator Trap
Courtesy, Illinois Engineering Company, Chicago

with a vertical self-cleaning seat; and Fig. 137 indicates the method used in draining the bottoms of down-feed risers or the ends of mains.

Another form of this valve, called a *water-seal motor*, which is used under practically the same conditions, is shown in Fig. 138.
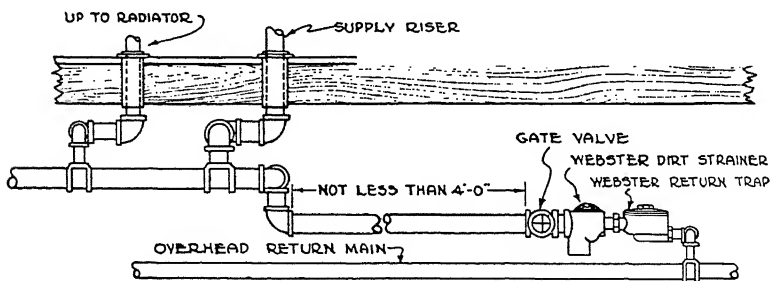

Fig. 137. Showing Method of Draining Bottoms of Down-Feed Risers or Ends of Mains

Its action is as follows: ordinarily, the Seal *A* is down, and the central tube-valve is resting upon the seat, closing the port *K* and preventing direct communication between the interior of the motor-

body E and the outlet L. The outlet is attached to a pipe leading
to a vacuum-pump, or other draining apparatus, which exhausts the
space F above the seal through the annular space between the spindle
B and the inside of the central tube G. The water of condensation,
accumulating in the radiator or coil, passes into the chamber E,
through the inlet C, rises in the chamber, and seals the space between
the seal-shell A and the sleeve of the bonnet D. The differential
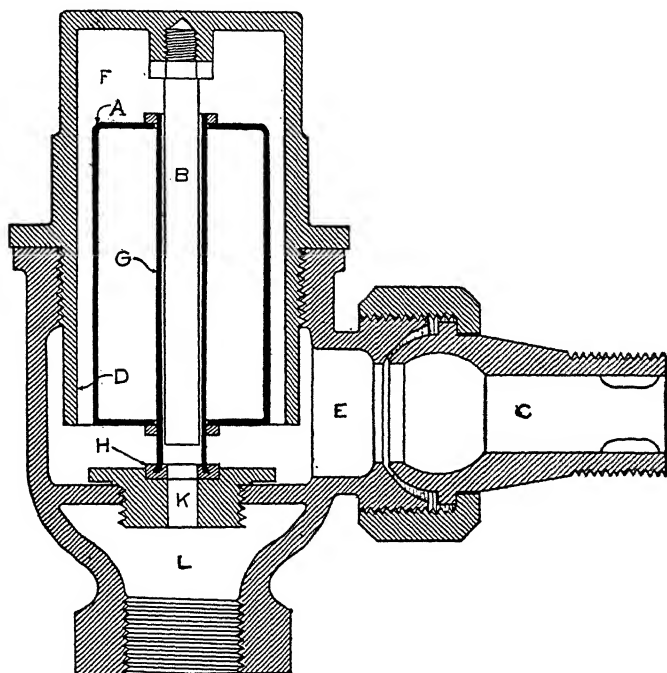
Fig. 138.   Water-Seal Motor

pressure thus created causes the seal A to rise, lifting the end of the
central tube off the seat, thus opening a clear passageway for the
ejection of the water of condensation.

When all the water of condensation has been drawn out of the
radiator, the seal and tube are reseated by gravity, thus closing the
port K, preventing waste or loss of steam; and the pressure is equal-
ized above and below the seal because of the absence of water. This
action is practically instantaneous. When the condensation is small
in quantity, the discharge is intermittent and rapid.

The space between the seal A and the sleeve of the bonnet D, and the annular space between the central tube G and the spindle B, form a passageway through which the air is continually withdrawn by the vacuum pump or other draining apparatus. This action con-
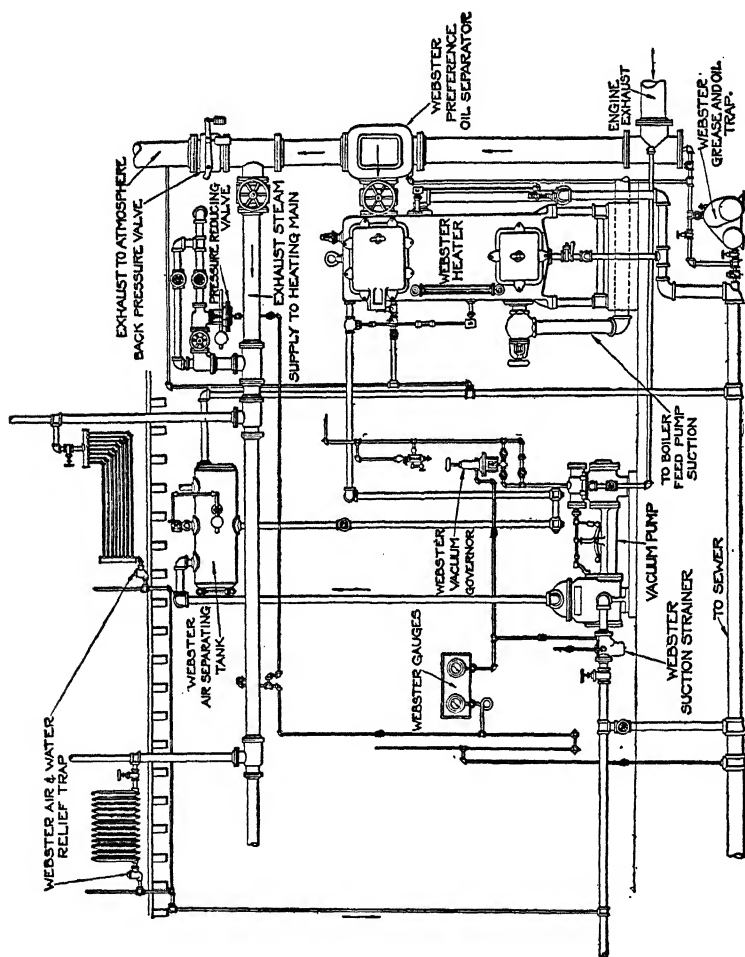

Fig. 139. Typical Layout of Webster Vacuum Heating System
Courtesy of Warren Webster and Company Camden, New Jersey

tinues as long as water is present. No adjustment whatever is necessary; the motor is entirely automatic.

One special advantage claimed for this system is that the amount of steam admitted to the radiators may be regulated to suit the

requirements of outside temperature, and is possible without water-logging or hammering. This may be done at will by closing down on the inlet supply to the desired degree. The result is the admission of a smaller amount of steam to the radiator than it is calculated to condense normally. The condensation is removed as fast as formed by the opening of the thermostatic valve.

The general application of this system to exhaust heating is shown in Fig. 139. Exhaust steam is brought from the engine as shown; one branch leads outboard through a back-pressure valve, while the other connects with the heating system through a grease extractor. A live steam connection is made through a reducing valve, as in the ordinary system. Valved connections are made with the coils and radiators in the usual manner; but the return valves are replaced by the special thermostatic valves.

The main return is brought down to a vacuum pump which discharges into a return tank (not shown in the illustration), where the air is separated from the water and passes off through a vapor pipe at the top. The condensation then flows into the feed-water heater, or receiving tank, from which it is automatically pumped back into the boilers. The cold-water feed supply is connected with the return tank, and a small cold-water jet is connected into the suction at the vacuum pump for increasing the vacuum in the heating system by the condensation of steam at this point.

Paul System. In the Paul system the suction is connected with the air-valves instead of with the returns, and the vacuum is produced by means of a steam ejector or a pump. The returns are carried back to a receiving tank and pumped back to the boiler in the usual manner. The ejector in this case is called the *exhauster*.

The general method of making the pipe connections with the radiators in this system is shown in Fig. 140; and the details of connection at the exhauster are shown in Fig. 141. A A are the returns from the air-valves and connect with the exhausters, as shown. Live steam is admitted in small quantities through the valves BB; and the mixture of air and steam is discharged outboard through the pipe C. D D are gauges showing the pressure in the system; and E E are check-valves. The advantage of this system depends principally upon the quick removal of air from the various radiators and pipes, which constitutes the principal obstruction to circulation;

the inductive action in many cases is sufficient to cause the system
to operate somewhat below atmospheric pressure.

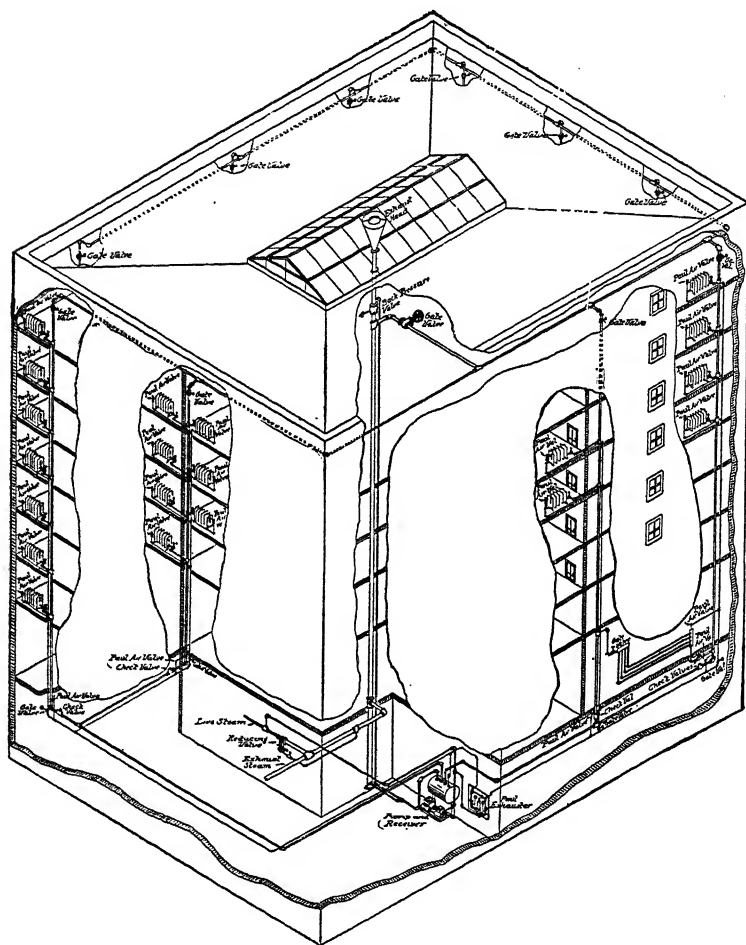Fig. 140.   Showing General Method of Making Pipe and Radiator Connections in
Paul System

Where exhaust steam is used for heating, the radiators should
be somewhat increased in size, owing to the lower temperature of
the steam.   It is common practice to add from 20 to 30 per cent to
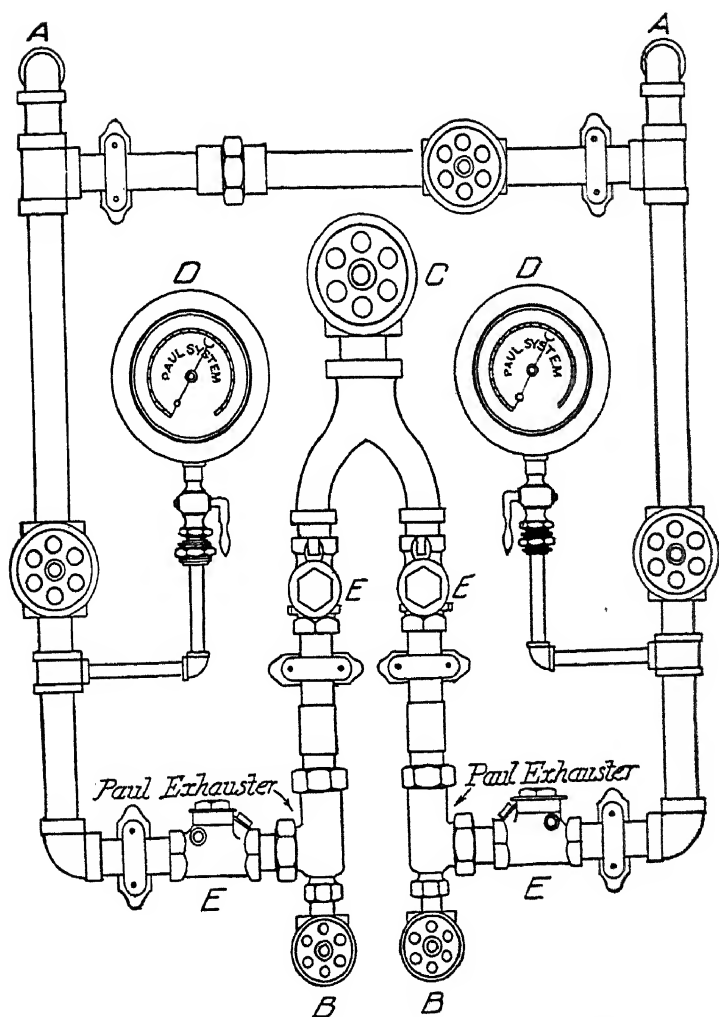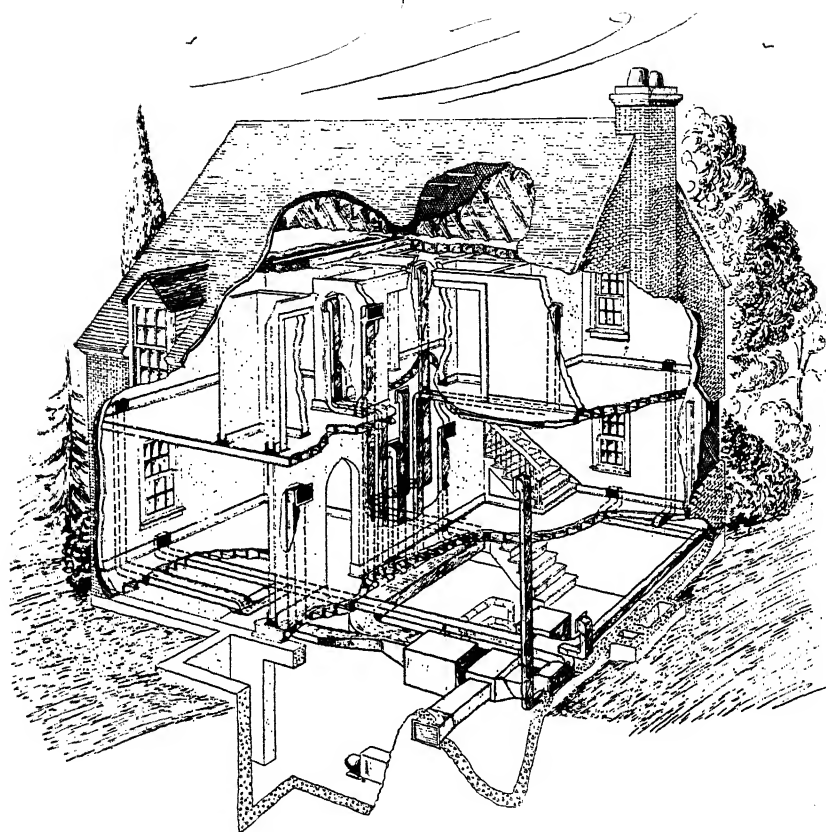the sizes required for low pressure live steam.

Fig. 141. Details of Connections at Exhauster, Paul System

CUTAWAY SHOWING TYPICAL DUCT SYSTEM

CHAPTER XIV

*FANS

There are two classes of fans, (1) disk or propeller, and (2) centrifugal. The disk fans discharge air in the direction of the shaft axis, or axially, and the centrifugal fans have a radial flow of air.

Centrifugal Fans. Centrifugal fans may be divided into two classes (1) those with straight radial tips and (2) those having rotors with blades curved in reference to their direction of motion.

In any centrifugal fan there are two sources of pressure: (1) pure centrifugal force due to the rotation of an enclosed column of air, (2) kinetic energy developed in the air by its velocity upon leaving the periphery of the fan rotor. The amount of centrifugal force given the air depends largely upon the ratio of the tangential velocity of the air leaving the periphery of the rotor to the velocity of the air entering the fan at the heel of the blades.

When the flow of air through the rotor of a fan is partially obstructed, the centrifugal effect in the rotor produces a compression known as "static pressure." On the other hand, the kinetic energy of the air leaving the periphery of the rotor must be converted largely into potential energy in the form of static pressure before being serviceable. This conversion from kinetic energy of velocity into static pressure is ordinarily accomplished in the scroll formation of the fan housing. A still further conversion is often secured, where the velocity leaving the outlet is high, by means of a diverging nozzle on the outlet of the fan.

The velocity of the air and its corresponding pressure when leaving the tip of the blades is greatly in excess of that ordinarily required in the piping system, but the static pressure is too low. By enclosing the wheel in a casing having a properly designed scroll, this velocity is reduced, and a part of the velocity pressure is converted to static pressure. Since the static pressure, due to the wheel, varies as the difference of the squares of the rotational velocities at the periphery and inlet, it is evident that the shorter the fan blade, the greater

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*All tables shown in this chapter are greatly reduced. Manufacturers will supply complete tables upon request.

must be the dependence on the scroll-shaped housing to obtain the desired static pressure. For this reason, the proper design of the housing is of greater importance in a short blade Multi-Vane type of fan than in the case of the older styles.

The short curved blade fans with backward bent blades give the highest static efficiencies. This is to be expected, since the fan with the backward sloping blade runs at a higher speed to obtain the same
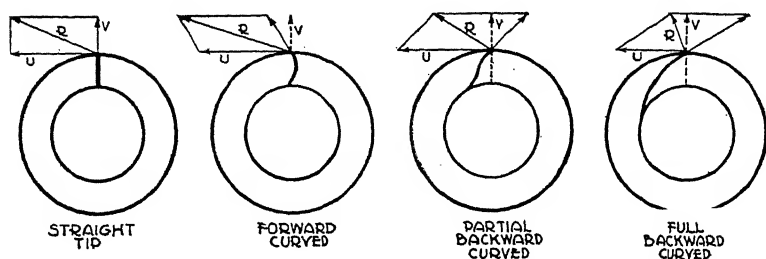

Fig. 142. Velocity Diagrams of Principal Wheels. U=Tip Speed; V=Radial Velocity of the Air; R=Actual Velocity with Respect to the Fan Casing

pressure and capacity and consequently develops more centrifugal pressure than another fan of the same size.

A sketch of the velocity diagrams of the principal wheels is shown in Fig. 142. The radial or straight tip type of wheel, whose performance is shown on PD 8890 (Fig. 143), has fairly steep pressure and efficiency curves with a steadily increasing straight-line horsepower. The maximum efficiency is at a point slightly to the right of the point where the pressure begins to drop.

The forward curved type of wheel, whose performance is shown on PD 8889 (Fig. 144), has an air velocity greater than the rotational speed of the wheel. This condition results in increase of the pressure until a point is reached at which the velocity through the wheel is so high that the air can follow the blade no longer, and an unstable condition is produced. However, when the velocity has passed this point, the space between the blades fills up and the fan operates in a stable condition. The maximum efficiency occurs where the pressure just begins to decrease and the horsepower increases rapidly with an increase in volume.

The partial backward curved type of wheel, whose performance is shown on PD 8891 (Fig. 145), has a steeper pressure curve than
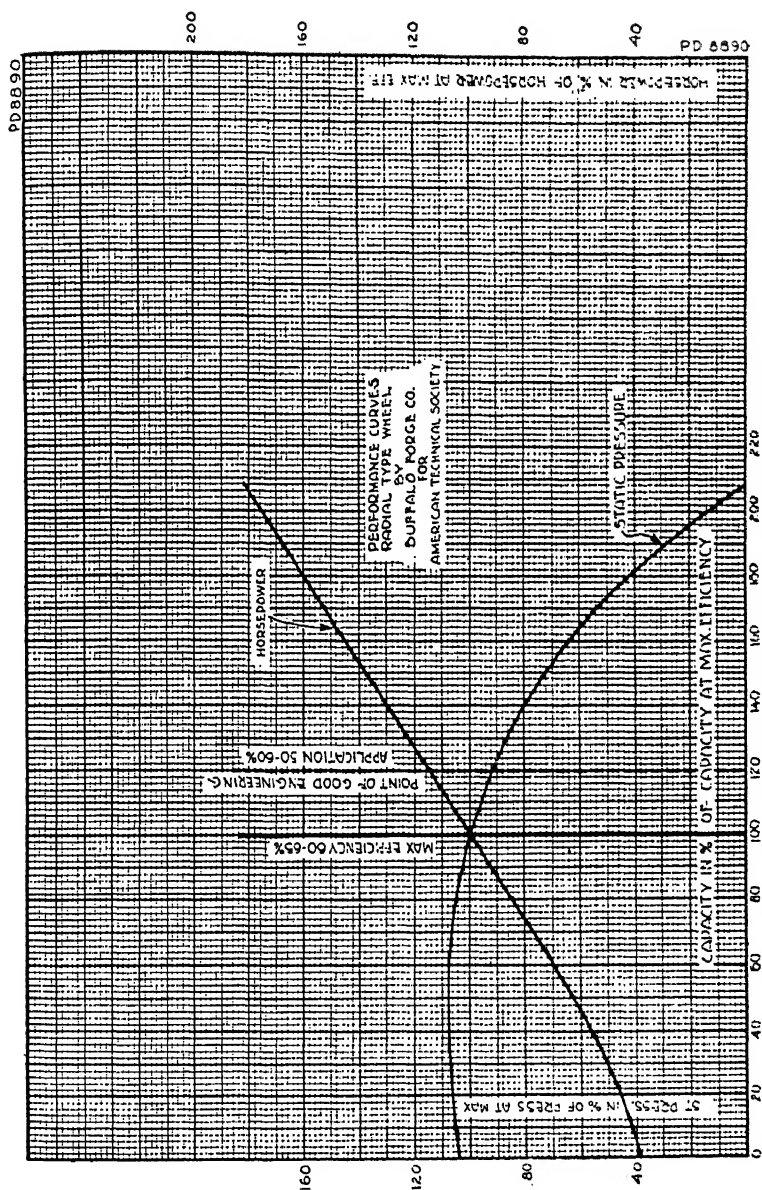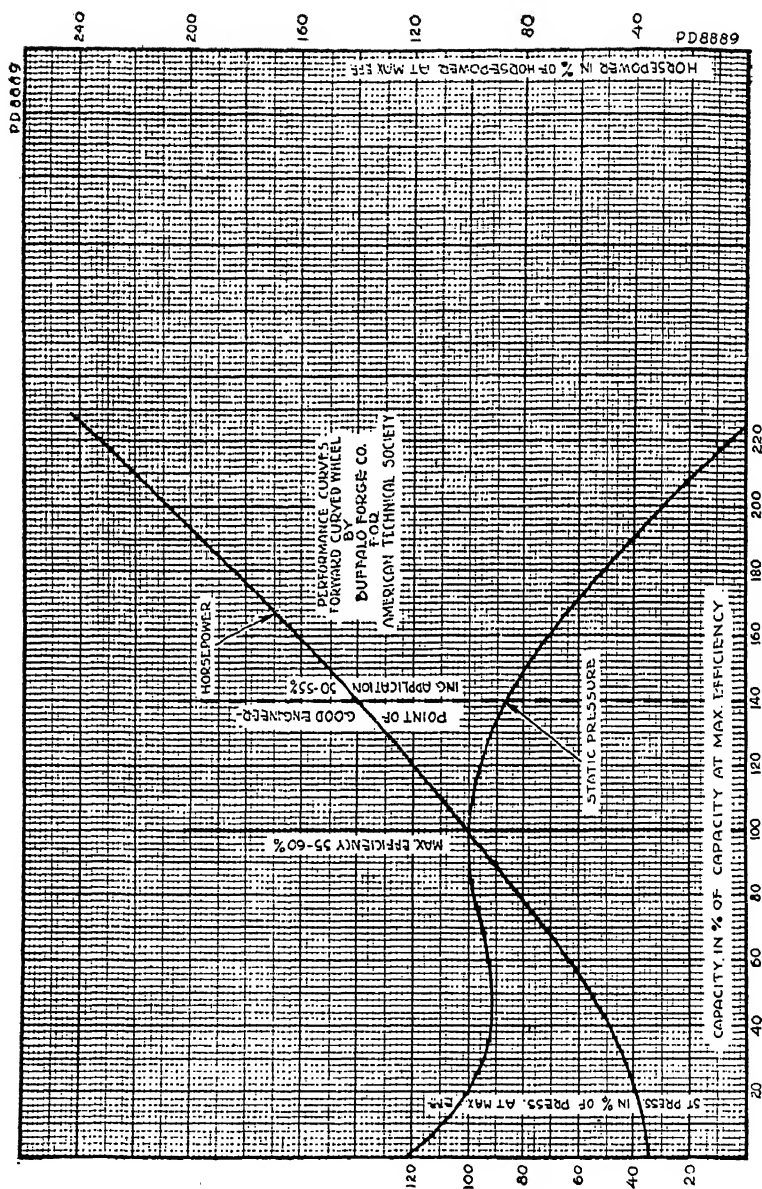
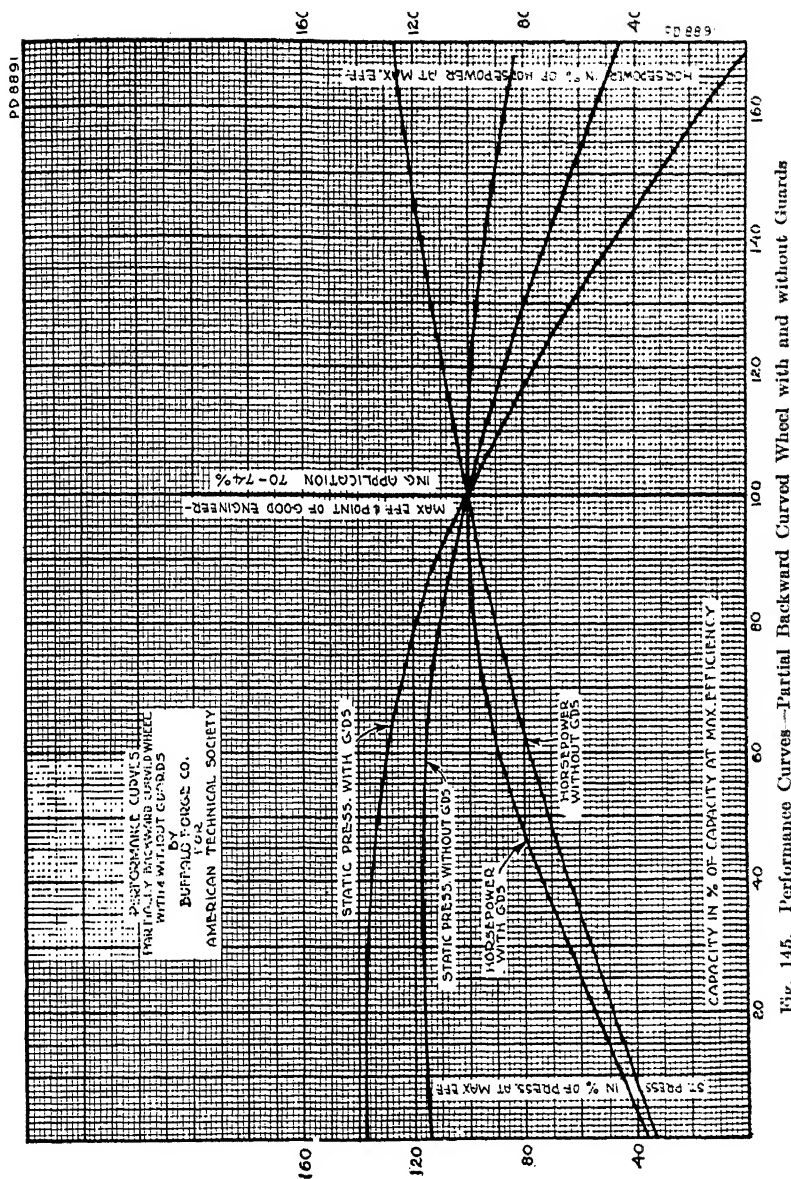Fig. 144.   Performance Curves—Forward Curved Wheel

Fig. 145. Performance Curves.—Partial Backward Curved Wheel with and without Guards

either the radial or the forward curved type of fan. The maximum efficiency occurs where the pressure begins to drop off rapidly and the horsepower of the partial backward curved type increases with an increase in volume. However, it will be shown later that the use of inlet vanes increases the efficiency and makes the horsepower self-limit-loading.

The full backward curved type of wheel has pressure and efficiency characteristics similar to those of the partial backward curved wheel, but the horsepower characteristic is of the self-limit-loading or non-overloading type. The performance of this type of fan with the full backward curved wheel is shown on PD 8892 (Fig. 146). This last type of fan has to run faster than the partial backward curved type which is a reason for the preference of a partially curved fan, using the inlet guards, to a fully curved fan.

The straight tip type of wheel is the best type for conveying or planing mill exhaust, because it has less resistance to the passage of materials through the blades than other types of fans. The fans of this type are the PMX (Fig. 147) and the SSPMX (Fig. 148) which are of the steel plate type, and the volume fan which has a cast-iron housing. Tables 64 and 65 cover these fans.

There is also the high pressure and low capacity type such as the blowers "E" (Fig. 149) and "RE" (Fig. 150). Tables 66 and 67 deal with these blowers.

The straight tip type is also used for induced draft work. An example of this type is the SSL (Fig. 151). The use of this type of blade for induced draft work will be discussed later.

The forward curved type may be satisfactorily operated if the rating point is well to the right of the peak of the pressure curve. If, however, the fan is rated near the peak or to the left of the peak in the dip, the fan will perform erratically because at one pressure it may deliver three different capacities. To avoid this, the fan must be rated to the right of the peak. However, when going to the right the efficiency drops off. If a low speed is desired and efficiency is not a prime factor, this type of fan may be used.

Because of the "hunting" characteristic of the forward curved fan, it is not satisfactory for forced draft work where a steady pressure is desired. Forced draft fans should have a large pressure reserve. For example, if a fire becomes densely packed or clinkers form, the
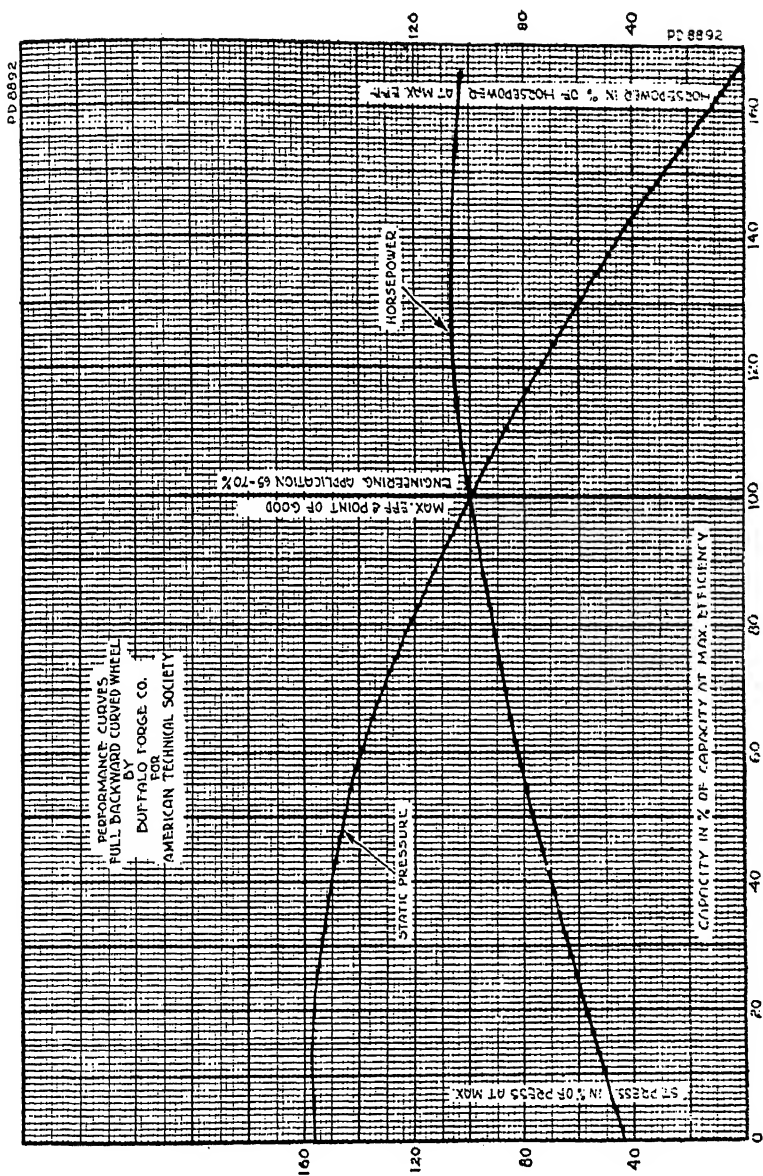
Fig. 146. Performance Curves—Full Backward Curved Wheel
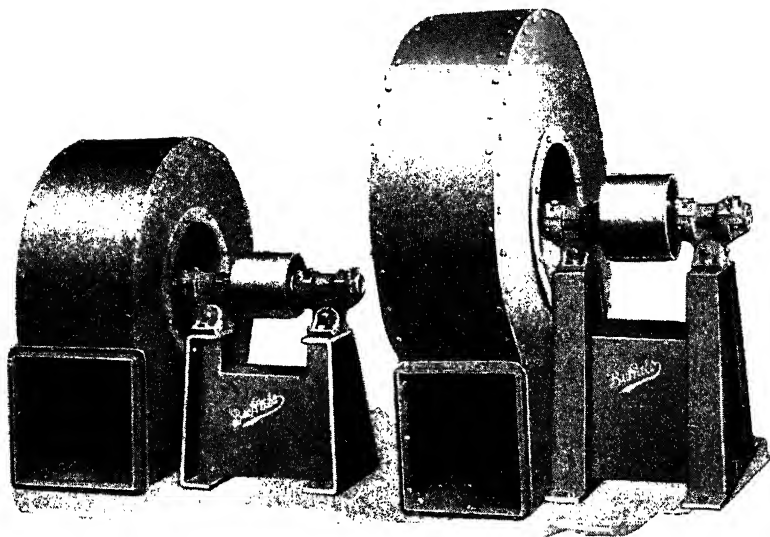
Fig. 147.   Buffalo Type PMX Fan   Fig. 148.   Buffalo Type SSPMX Fan
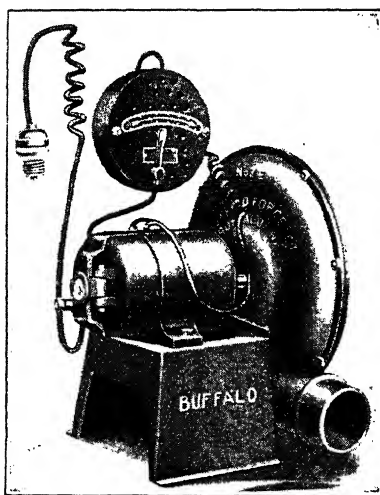*Courtesy of Buffalo Forge Company, Buffalo, N. Y.*

Fig. 149.   Buffalo Type E Blower
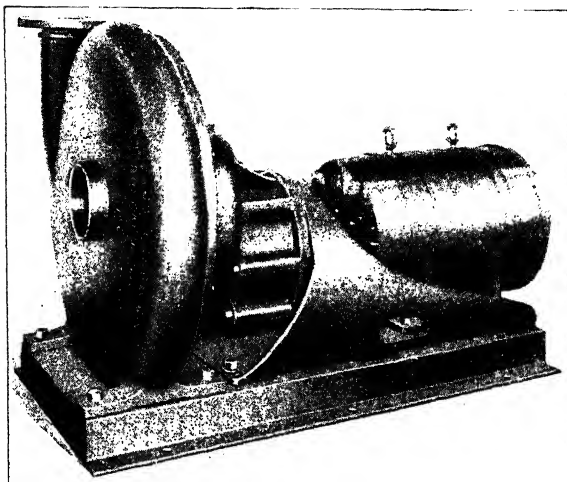*Courtesy of Buffalo Forge Company,*
Buffalo, N. Y.

Fig. 150. Buffalo Type RE Blower
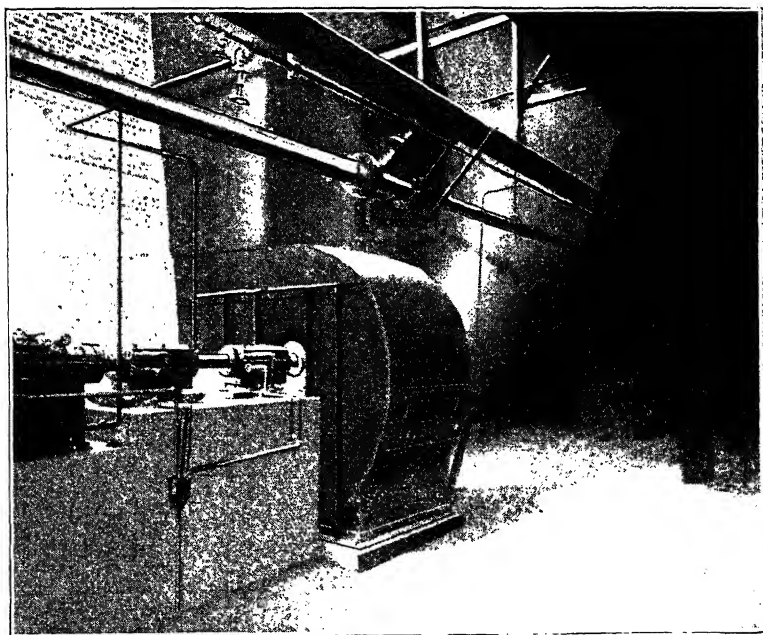*Courtesy of Buffalo Forge Company, Buffalo, N. Y.*

Fig. 151. Buffalo Type SSL Fan
Courtesy of Buffalo Forge Company, Buffalo, N. Y.

Table 64. *Standard Mill Exhausters (Type PMX)

Size 35—Capacity and Static Pressure at 70°F. and 29.92″ Barometer

Outlet Velocity Ft. per Min.	Capacity Cu.Ft. Air per Min.	1″ S.P.		2″ S.P.		3″ S.P.		4″ S.P.		5″ S.P.		6″ S.P.		7″ S.P.		8″ S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
1000	1067	600	.27	834	.58	1024	.94	1180	1.26	1324	1.72	1448	2.14	1564	2.61	1674	3.09
1250	1335	610	.34	840	.70	1026	1.09	1182	1.53	1326	1.98	1449	2.46	1566	2.95	1676	3.46
1500	1600	624	.41	846	.82	1030	1.26	1184	1.74	1328	2.23	1450	2.77	1568	3.31	1678	3.87
1750	1870	644	.51	854	.94	1032	1.43	1186	1.96	1330	2.48	1451	3.09	1570	3.68	1680	4.30
2000	2134	666	.61	872	1.09	1042	1.63	1192	2.19	1332	2.80	1452	3.42	1572	4.04	1682	4.71
2250	2400	692	.73	892	1.26	1058	1.84	1200	2.44	1338	3.09	1458	3.74	1574	4.43	1684	5.14
2500	2670	724	.89	912	1.46	1072	2.06	1216	2.71	1344	3.40	1462	4.10	1576	4.84	1686	5.58
2750	2935	754	1.05	938	1.67	1086	2.33	1230	3.00	1350	3.72	1472	4.47	1584	5.28	1688	6.05
3000	3201			960	1.87	1114	2.61	1244	3.33	1370	4.08	1484	4.88	1588	5.70	1694	6.55
3250	3470			988	2.16	1134	2.93	1266	3.68	1386	4.48	1500	5.30	1600	6.16	1702	7.04
3500	3740			1018	2.45	1158	3.25	1286	4.07	1400	4.91	1514	5.75	1614	6.68	1712	7.59
3750	4000			1046	2.80	1180	3.62	1310	4.48	1426	5.39	1532	6.30	1630	7.23	1728	8.18
4000	4268			1080	3.11	1212	3.98	1334	4.91	1444	5.87	1546	6.82	1646	7.80	1744	8.78
4500	4800					1272	4.95	1384	5.90	1496	6.92	1596	7.98	1690	9.06	1774	10.2
5000	5335							1446	7.08	1542	8.15	1642	9.32	1734	10.5	1824	11.8
5500	5870							1506	8.38	1604	9.58	1700	10.8	1784	12.1	1870	13.5
6000	6402									1666	11.2	1754	12.5	1838	13.8	1922	15.2
6500	6940											1816	14.5	1894	15.9	1974	17.2
7000	7480											1882	16.4	1956	18.3	2034	19.7

*Courtesy of Buffalo Forge Company.

Table 65. *Slow Speed Mill Exhausters (Type SSPMX)

Size 35—Capacity and Static Pressure at 70°F. and 29.92″ Barometer

Outlet Velocity Ft. per Min.	Capacity Cu.Ft. Air per Min.	1″ S.P.		2″ S.P.		3″ S.P.		4″ S.P.		5″ S.P.		6″ S.P.		7″ S.P.		8″ S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
1000	1069	414	.25														
1250	1338	415	.30	586	.64												
1500	1605	418	.34	586	.73	717	1.16										
1750	1870	427	.41	587	.82	717	1.29	828	1.81								
2000	2140	439	.49	590	.92	717	1.43	828	1.98	925	2.56						
2250	2405	451	.57	597	1.04	719	1.57	828	2.15	925	2.78	1014	3.42	1097	4.10		
2500	2670	467	.67	607	1.17	724	1.73	830	2.34	925	3.00	1014	3.69	1097	4.40		
2750	2940			619	1.32	730	1.91	833	2.55	926	3.24	1014	3.96	1097	4.71	1169	5.50
3000	3210			631	1.49	741	2.11	837	2.77	930	3.48	1015	4.24	1097	5.02	1169	5.83
3250	3475			644	1.66	752	2.31	846	3.00	935	3.75	1018	4.53	1098	5.35	1169	6.19
3500	3740			659	1.86	764	2.55	855	3.28	942	4.05	1022	4.85	1100	5.70	1171	6.58
3750	4010					775	2.81	867	3.56	951	4.36	1029	5.20	1103	6.05	1174	6.98
4000	4275			-		789	3.08	879	3.88	962	4.72	1038	5.58	1110	6.47	1179	7.38
4500	4810					820	3.69	905	4.58	985	5.48	1058	6.40	1125	7.35	1191	8.35
5000	5350							934	5.36	1012	6.32	1080	7.32	1149	8.35	1210	9.38
5500	5880									1040	7.30	1108	8.40	1171	9.45	1231	10.5
6000	6420											1135	9.50	1195	10.7	1258	11.9
6500	6950													1227	12.1	1285	13.3
7000	7480																

*Courtesy of Buffalo Forge Company.

Table 66. Multi-Rating Tables

Buffalo Type "E" Blowers
Capacities and Static Pressures at 70°F. and 29.92" Barometer

No. 2E Blower 1750 R.P.M.			No. 2EH Blower 3450 R.P.M.			No. 3E Blower 3450 R.P.M.			No. 4E Blower 3450 R.P.M.		
Cap.	Static	Hp.	Cap.	Static	Hp.	Cap.	Static	Hp.	Cap.	Static	Hp.
30	.821	.010	30	3.35	.060	50	3.35	.072	100	3.65	.150
50	.730	.014	60	3.19	.078	100	3.24	.101	150	3.58	.175
70	.593	.018	100	2.84	.108	150	2.98	.140	200	3.46	.210
130	.0	.027	150	2.13	.140	200	2.60	.177	250	3.30	.250
			200	1.30	.171	250	2.02	.216	300	3.05	.290
			250	.45	.200	300	1.33	.255	350	2.70	.330
						350	.55	.292	400	2.25	.375
									450	1.75	.415

No. 4½ E

	1750 R.P.M.								3450 R.P.M.							
Cap. C.F.M.	12½" Wheel		11¾" Wheel		11" Wheel		9" Wheel		12½" Wheel		11¾" Wheel		11" Wheel		9" Wheel	
	Static	Hp.	Static	Hp.	Static	Hp.	Static	Hp.	Static	Hp.	Static	Hp.	Static	Hp.	Static	Hp.
50	2.72	.065	2.30	.058	2.02	.049	1.30	.032	10.55	.50	8.92	.44	7.80	.38	5.12	.25
100	2.70	.085	2.28	.075	2.00	.064	1.27	.045								
150	2.65	.109	2.24	.096	1.96	.082	1.21	.060								
200	2.56	.132	2.16	.117	1.88	.100	1.12	.075	10.51	.65	8.90	.58	7.80	.50	5.08	.35
250	2.43	.155	2.04	.138	1.75	.120	.97	.090								
300	2.26	.179	1.86	.160	1.55	.139	.76	.104	10.30	.84	8.71	.75	7.59	.64	4.75	.47

Table 67. Buffalo Type Blowers
3450 R.P.M.

Cap. C.F.M.	No. 2 RE						Cap. C.F.M.	No. 3 RE					
	18¾" Wheel		17½" Wheel		16½" Wheel			21½" Wheel		20½" Wheel		18½" Wheel	
	Static	Hp.	Static	Hp.	Static	Hp.		Static	Hp.	Static	Hp.	Static	Hp.
50	21.6	.89	19.8	.76	17.6	.64	100	30.5	2.00	27.3	1.70	22.2	1.20
100	21.1	.98	19.1	.82	17.0	.71	200	30.8	2.42	27.7	2.10	22.5	1.55
150	20.4	1.13	18.4	.97	16.3	.72	300	30.9	2.90	27.7	2.52	22.3	1.95
200	19.4	1.35	17.3	1.15	15.0	.97	400	30.5	3.45	27.2	3.02	21.5	2.38
250	17.4	1.60	15.0	1.39	12.7	1.16	500	29.4	4.00	25.5	3.52	19.5	2.82
300	14.2	1.87	11.9	1.65	9.5	1.37	600	26.7	4.55	22.6	4.03	16.7	3.28
350	10.5	2.17	8.0	1.91	5.6	1.60	700	23.2	5.18	19.3	4.55	13.2	3.70
400	6.0	2.47	3.6	2.20	1.0	1.82	800	19.2	5.80	15.3	5.08	9.4	4.12

resistance to the air flow increases. Unless the fan has a steep pressure curve—like that in the partial backward and full backward curved blades—the capacity will be decreased, with a resultant decrease in boiler efficiency.

In addition, forward curved fans are not suitable for parallel operation because one fan may take most of the load and result in the burning out of one or both motors. The forward curved fan with its

Table 68. Buffalo Limit Load Conoidal Fan (Type CL)

Capacities and Static Pressures of Single Width at 70°F. and 29.92″ Barometer

Size 6—Wheel diameter 36″. Wheel width 16½″. Approximate weight single fan 1000 lbs.

Outlet Velocity Feet per Minute	Capacity Cubic Feet of Air per Minute	¼″ S.P.		⅜″ S.P.		½″ S.P.		⅝″ S.P.		¾″ S.P.		⅞″ S.P.		1″ S.P.		Limit Load	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
800	5,820	279	.348	317	.486	352	.636	384	.796	415	.960	441	1.13	468	1.31	250	.255
900	6,547	296	.416	330	.564	364	.725	394	.896	424	1.07	450	1.26	476	1.45	280	.358
1000	7,275	313	.499	346	.656	376	.825	405	1.01	433	1.19	460	1.39	485	1.59	310	.486
1100	8,002	332	.595	362	.764	390	.940	418	1.13	445	1.33	470	1.54	495	1.75	340	.641
1200	8,730	351	.699	379	.885	406	1.07	431	1.27	457	1.48	482	1.70	505	1.92	370	.826
1300	9,457	370	.825	397	1.02	422	1.22	447	1.43	470	1.64	494	1.87	518	2.11	400	1.04
1400	10,185	390	.966	416	1.17	440	1.39	464	1.61	485	1.83	508	2.07	531	2.30	430	1.30
1500	10,912	410	1.12	435	1.34	459	1.57	481	1.80	501	2.04	523	2.28	544	2.53	460	1.59
1600	11,640	430	1.30	455	1.53	478	1.77	499	2.02	519	2.27	539	2.52	559	2.77	490	1.92
1700	12,367	451	1.49	475	1.75	497	1.99	517	2.25	537	2.52	556	2.78	575	3.05	520	2.30
1800	13,095	472	1.72	495	1.97	516	2.23	536	2.50	555	2.78	573	3.06	592	3.34	550	2.72
1900	13,822	493	1.95	515	2.22	535	2.50	555	2.78	573	3.07	590	3.36	609	3.65	580	3.19
2000	14,550	515	2.18	535	2.50	555	2.79	574	3.08	592	3.37	609	3.68	627	3.99	610	3.71
2100	15,277			555	2.80	575	3.10	594	3.41	611	3.72	628	4.03	645	4.36	640	4.28
2200	16,005			576	3.17	595	3.44	614	3.76	630	4.09	647	4.42	664	4.74	670	4.91

Size 6½—Wheel diameter 39″. Wheel width 17¹³⁄₁₆″. Approx. wt. of single fan 1200 lbs.

Outlet Velocity Feet per Minute	Capacity Cubic Feet of Air per Minute	¼″ S.P.		⅜″ S.P.		½″ S.P.		⅝″ S.P.		¾″ S.P.		⅞″ S.P.		1″ S.P.		Limit Load	
800	6,830	258	.41	292	.57	324	.75	354	.94	384	1.13	408	1.33	432	1.53	220	.27
900	7,684	272	.49	306	.66	336	.85	364	1.05	392	1.26	416	1.48	440	1.70	250	.39
1000	8,538	288	.59	320	.77	348	.97	374	1.18	400	1.40	424	1.63	448	1.87	280	.54
1100	9,392	306	.70	334	.90	360	1.10	386	1.33	410	1.56	434	1.81	456	2.05	310	.74
1200	10,246	324	.82	350	1.04	374	1.26	398	1.49	422	1.74	444	1.99	466	2.25	340	.97
1300	11,100	342	.97	366	1.20	390	1.44	412	1.68	434	1.93	456	2.20	478	2.47	370	1.25
1400	11,953	360	1.13	384	1.37	406	1.63	428	1.89	448	2.15	468	2.42	490	2.70	400	1.58
1500	12,807	378	1.32	402	1.58	424	1.84	444	2.11	462	2.40	482	2.68	502	2.98	430	1.97
1600	13,661	396	1.53	420	1.80	442	2.07	460	2.37	478	2.65	498	2.95	516	3.25	460	2.41
1700	14,515	416	1.75	438	2.05	458	2.34	476	2.65	496	2.95	514	3.28	530	3.58	490	2.92
1800	15,368	436	2.02	456	2.31	476	2.63	494	2.93	512	3.25	530	3.60	546	3.93	520	3.48
1900	16,222	456	2.29	474	2.60	494	2.93	512	3.25	528	3.60	546	3.95	562	4.30	550	4.13
2000	17,076	476	2.55	492	2.93	512	3.28	530	3.60	546	3.95	562	4.33	578	4.68	580	4.83
2100	17,930			512	3.28	530	3.65	548	4.00	564	4.35	580	4.73	594	5.13	610	5.62
2200	18,784			532	3.73	548	4.03	566	4.43	582	4.80	598	5.20	612	5.58	640	6.50

Size 10—Wheel diameter 60″. Wheel width 27⅜″. Approx. wt. of single fan 2840 lbs.

Outlet Velocity Feet per Minute	Capacity Cubic Feet of Air per Minute	¼″ S.P.		⅜″ S.P.		½″ S.P.		⅝″ S.P.		¾″ S.P.		⅞″ S.P.		1″ S.P.		Limit Load	
800	16,167	168	.965	190	1.35	211	1.77	230	2.21	249	2.67	265	3.14	281	3.62	150	.708
900	18,187	178	1.16	198	1.57	218	2.01	236	2.49	254	2.98	270	3.49	286	4.01	170	1.03
1000	20,208	188	1.39	207	1.82	226	2.29	243	2.80	260	3.32	276	3.86	291	4.42	190	1.44
1100	22,229	199	1.65	217	2.12	234	2.61	251	3.14	267	3.69	282	4.26	297	4.86	210	1.95
1200	24,250	210	1.94	227	2.46	243	2.98	259	3.53	274	4.11	289	4.71	303	5.33	230	2.55
1300	26,271	222	2.29	238	2.83	253	3.39	268	3.97	282	4.57	296	5.19	311	5.85	250	3.28
1400	28,292	234	2.68	249	3.25	264	3.85	278	4.46	291	5.08	305	5.74	319	6.38	270	4.14
1500	30,312	246	3.11	261	3.73	275	4.35	288	4.99	301	5.67	314	6.33	327	7.03	290	5.12
1600	32,333	258	3.61	273	4.25	286	4.91	299	5.60	311	6.29	324	6.99	335	7.70	310	6.26
1700	34,354	270	4.14	285	4.85	297	5.53	310	6.24	322	6.99	334	7.72	345	8.47	330	7.55
1800	36,375	283	4.78	297	5.47	309	6.19	322	6.94	333	7.70	344	8.50	355	9.28	350	9.00
1900	38,396	296	5.41	309	6.17	321	6.93	333	7.72	344	8.52	354	9.33	365	10.1	370	10.7
2000	40,417	309	6.05	321	6.93	333	7.75	344	8.54	355	9.35	365	10.2	376	11.1	390	12.5
2100	42,437			333	7.78	345	8.61	356	9.48	366	10.3	377	11.2	387	12.2	410	14.5
2200	44,458			346	8.80	357	9.54	368	10.5	377	11.4	388	12.3	398	13.2	430	16.7

Note: For double width fans delivering double the above capacities, multiply outlet velocity by 1.11; R.P.M. by 1.01; Hp. by 2.04.

Actual fan horsepowers are shown in table, add for loss through drive when selecting motor.

constantly increasing horsepower provides no guarantee that the motor will not be overloaded due to changing conditions, as does a fan of the SL and CL type, Figs. 152 and 153. See Table 68 for ratings.

The partial backward curved type—such as the CL for ventilating and the SL for forced draft—when used with fixed inlet guards
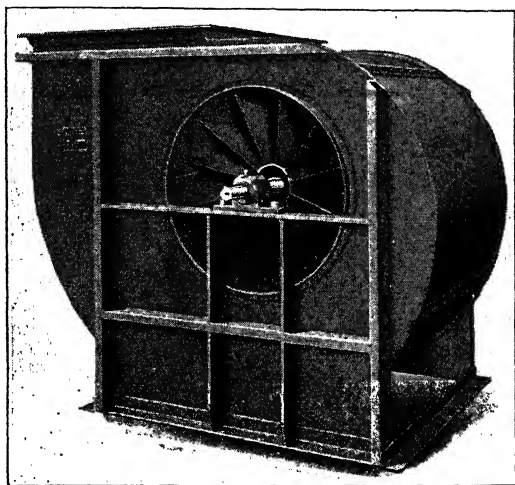

Fig. 152.   Buffalo Type SL Fan.   (Rating tables can be secured
from manufacturer.)
Courtesy of Buffalo Forge Company, Buffalo, N. Y.

provides a limit load fan which has a lower speed than the full backward curved type. The inlet guards reduce impact or shock loss at the heel of the blade by turning the air in the direction of rotation of the fan. The comparison between a fan without inlet guards and one with inlet guards is shown on PD 8891 (Fig. 145).

These stationary guards or vanes serve as mechanical guards and as air straighteners when inlet boxes are attached to the fan housing. In this way they eliminate the disturbing and uneven flow caused by some duct connections. Moreover, tests show that although the use of inlet vanes requires higher tip speeds for a given capacity and pressure, they reduce the sound. Tests made with a limit load conoidal ventilating fan, equipped with inlet vanes, and a commercial forward curved blade fan show that for a given capacity and static pressure, although the tip speed of the former is considerably in excess of the latter, the sound intensity measured in both the fan inlet and

outlet is less. The reduction in inlet noise is due partly to the sound absorbing qualities of the vanes themselves and partly to the reduction of shock loss at the inlet edge of the fan blade. But the reduction in noise on the discharge side can be due only to a more efficient and uniform flow leaving the fan rotor and fan discharge.

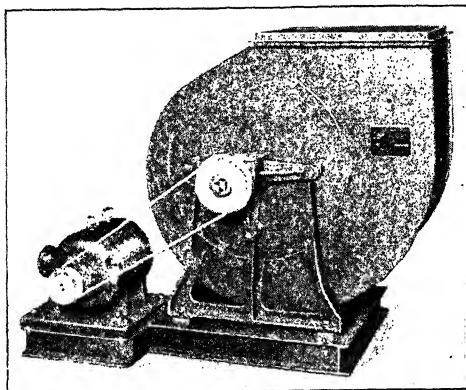The full backward curved type is used in heating and conditioning units where the limit-load feature without guards is desired. This

Fig. 153. Buffalo Type CL Fan
Courtesy of Buffalo Forge Company, Buffalo, N. Y.

type requires a higher speed motor than a forward curved wheel type requires.

The choice of an induced draft fan must be influenced by the part that erosion will play in the life of the fan. Backward curved or straight tip blades are preferable to forward curved blades.

Disk or Propeller Fans. The disk or propeller fans are primarily displacement fans and are frequently found where air is to be moved without the aid of piping. They normally operate at free delivery or against moderate pressures and are often installed in an outside wall to exhaust foul air or to supply fresh air from without. They are also used for recirculation of air within a room or chamber, as in drying rooms, or behind the coils of a unit heater. Since effective temperature may be lowered by air motion, these fans give a cooling effect in the summer time. For this purpose the fan is usually of the portable type and is placed upon a wall, stand, or desk. Probably the most familiar use of this type of fan is in the automobile, where the fan is used for cooling the radiator water.

Disk and propeller fans discharge the air axially. The former have flat blades while the latter have curved blades, usually die-formed. When proportioning as to size, the fans must be of similar blade shape. Commercial disk fans usually have their blades set at an angle of from 20 to 30 degrees from the plane of rotation. The larger angle develops more pressure at the same speed and capacity but requires greater horsepower. Likewise, an increase in the number of blades tends to build up a greater pressure. However, when the blades overlap each other, there is an added surface resistance which is felt at the larger capacities and lower pressures. Where the fan operates against low static pressure, which approaches free delivery conditions, two to four blades at the steeper angles will be sufficient.

Disk and propeller fans usually are mounted in a simple ring frame which acts as a guard and facilitates the attachment of the unit. Where fans are operated against pressure, the location of the wheel in relation to the frame modifies the fan characteristics. The efficiency of the common axial blade fan is not as high as the centrifugal type, but enclosing it in a housing so the relative velocity of the air through the wheel is increased for a given speed and pressure, increases its efficiency. This means installing a long Venturi type of housing or a spiral discharge volute, which increases space and cost. Many claims have been made for the two-blade airplane type of propeller operating in a simple ring frame. In this type the acceleration of the air is caused by the blades themselves and the efficiency is not materially different from the conventional disk fan. It is true that propellers on airplanes show high efficiencies when advancing through air, but air flow through a propeller is different from air flow through the blades of a stationary fan. In other words, the flow lines are not the same, any more than are the flow lines through an orifice if the flow is reversed. For a free delivery condition the airplane type of propeller is frequently used where it is desirable to confine the air and project it a maximum distance.

The ordinary disk and propeller fans are widely used because of their low initial cost and small space requirement. For performance data on such fans, see Breezo Type, Fig. 154 and Table 69.

Fan Blade Erosion. The fine ash created by pulverized coal often causes rapid wearing away of induced-draft fan blades and parts of the fan scroll. Erosion varies approximately as the square of the

velocity of the abrasive particle of ash. Serious erosion occurs only
when flue-gas velocity over the blades of the blast wheel is high,
as it is in systems operating at high fan pressures. Abrasive quality
of the ash depends to some extent on the completeness of combustion.
Cases are known in which the fan pressures and velocities have been
about the same in two plants, but the blades of one fan were prac-
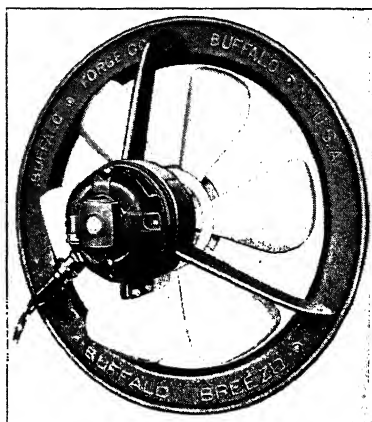tically free from erosion while the blades of the other showed con-

Fig. 154. BUFFALO Breezo Fan
Courtesy of Buffalo Forge Company, Buffalo, N. Y.

siderable wear. So far, no metal has proved more effective in resisting
this wear than ordinary steel plate.

To withstand erosion, wheel blades are sometimes provided with
heavy wearing plates where the greatest erosion occurs. Housings
are provided with easily renewable scroll liners.

The design of a fan blade is an important factor in its suscepti-
bility to erosion. Figs. 155 and 156 show the blades of the forward and
the backward curved types, with their corresponding parallelograms
of velocity. Both of these forms of blade are subject to powdered coal
ash wear, the greater wear occurring in the design that has the greater
velocity of gas at the blade tip.

The static pressure produced by a centrifugal fan equals the
pressure produced within the blade itself due to centrifugal force, plus
the static pressure converted in the volute housing by the velocity
of the air leaving the blade tip. The greater the forward angle at

the blade tip, the greater will be the leaving velocity and the greater the static pressure converted.

Let us compare the two blade types shown, each delivering the same capacity at the same pressure and having the same diameter wheels. In Figs. 155 and 156, P and P' represent the peripheral velocity of the tip of the blade, R and R' the radial velocity of the air through the blade, V and V' the resultant velocity of the air leaving the blade. The static pressure converted from this velocity
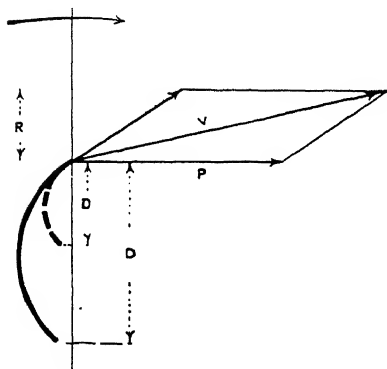

Fig. 155.   Forward Curved Blade (1)   Fig. 156.   Backward Curved Blade (2)
Courtesy of Buffalo Forge Company, Buffalo, N. Y.
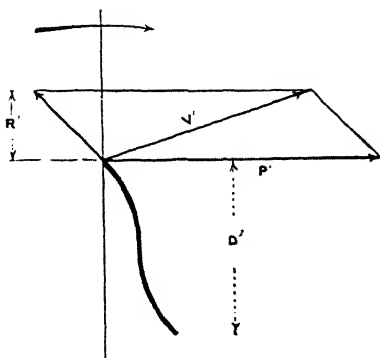
varies as the square of the velocity. The depth of blade D is the same as that of D'. At the same tip speed, the pressure produced within the blade due to centrifugal force would be the same.

The blade in Fig. 155 produces 1-inch static pressure at 2,700 feet per minute tip speed. Of this, 0.3 inch is produced within the blade by centrifugal force and 0.7 inch is converted from velocity V. The blade in Fig. 156 is operating at 4,000 feet per minute tip speed and, therefore, produces $\dfrac{4000^2}{2700} \times 0.3 = 0.66$ inch pressure within the blade. The relative velocity, V to V', is as 1 to 0.698, so that pressure converted from velocity V' will be $(0.698)^2 \times 0.7 = 0.34$ inch, making 1.0 inch static for the blade in Fig. 156.

The blade in Fig. 155 operates at lower speed than the blade in Fig. 156 on account of the higher velocity of the air at the tip of the blade. Conversely, the air in contact with the tip of the backward

curved blade is at lower velocity than the forward curved blade, although its actual tip speed is greater. Actually, depth D of the commercial forward curved blade is considerably less than the depth of the backward curved blade, so that the difference between V and V' will be even greater than shown in the example above.

Wear is greater on a forward-curved blade than on a backward curved blade in spite of its lower tip speed. Reference to the parallelograms of velocity shows that the air velocity and ash velocity along the blade tip are greater than in the case of the backward curved blade operating at the same capacity and pressure, and wear will be proportionately greater. For the same design of blade, the wear is greater when the tip speed is higher. The comparative wear on two different blade designs cannot be found by comparing tip speeds. Actual air velocities at blade tips must be compared.

The Buffalo SSL (Fig. 151) for induced draft has a forward curve at the heel of the blade but has a straight tip. This tends to reduce the operating speed and makes it easier to change the wheel size in case a slight change in fan speed is necessary.

Selecting Fans. When selecting a fan, first choose the type of fan which is best suited to the conditions. Then, from performance tables of that type of fan, select a fan which has the lowest horsepower or is operating at a motor speed, if direct connection is desired. If a basic table for any type of fan is available, by applying the laws of fan performance, the exact speed and horsepower for any condition and any size fan within the limits of operation of that type fan can be determined. All fan laws are based upon the fundamental theory that fan efficiencies remain constant, and when one or more conditions vary, the rest must vary accordingly.

The relationship between pressure, capacity, and horsepower is determined as follows:

Each cubic foot of air per minute moved against a total pressure of one inch water gauge (equivalent to 5.19 pounds per square foot) represents an energy expenditure of 5.19 foot pounds per minute or .000157 horsepower.

That is, with perfect efficiency it will require .000157 horsepower to move one cubic foot of air against one inch pressure. Therefore

$$Hp. = \frac{(.000157)\ (\text{c.f.m.})\ (\text{Total Pressure})}{\text{Total Efficiency}} \tag{31}$$

or

$$Hp. = \frac{(.000157)\ (\text{c.f.m.})\ (\text{Static Pressure})}{\text{Total Efficiency}} \tag{32}$$

Table 69*. Buffalo Breezo Propeller Fans

Size 12

Velocity through Wheel	Capacity Cu. Ft. Air per Min.	0" S.P.		⅛" S.P.		¼" S.P.		½" S.P.		¾" S.P.		1" S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
500	392	725	.004	1130	.020	1390	.042	1780	.100	2100	.177	2370	.261
600	471	870	.007	1240	.024	1490	.048	1868	.108	2178	.184	2448	.275
700	550	1015	.011	1355	.030	1593	.055	1960	.118	2258	.196	2525	.286
800	628	1160	.017	1473	.037	1700	.063	2055	.130	2340	.207	2605	.300
900	707	1305	.024	1593	.047	1810	.074	2160	.142	2435	.224	2689	.314
1000	785	1450	.033	1715	.058	1925	.086	2260	.156	2538	.241	2775	.335

Size 16

Velocity through Wheel	Capacity Cu. Ft. Air per Min.	0" S.P.		⅛" S.P.		¼" S.P.		½" S.P.		¾" S.P.		1" S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
500	698	543	.007	847	.034	1043	.075	1335	.177	1575	.313	1777	.464
600	837	652	.012	930	.041	1118	.085	1401	.192	1633	.327	1886	.489
700	977	761	.020	1016	.053	1194	.097	1470	.210	1693	.347	1893	.509
800	1116	870	.030	1105	.066	1275	.112	1540	.230	1757	.367	1954	.535
900	1256	978	.042	1195	.083	1358	.131	1620	.252	1826	.398	2016	.555
1000	1396	1088	.059	1286	.103	1443	.153	1695	.276	1903	.429	2081	.595

Size 18

Velocity through Wheel	Capacity Cu. Ft. Air per Min.	0" S.P.		⅛" S.P.		¼" S.P.		½" S.P.		¾" S.P.		1" S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
500	884	483	.009	753	.044	927	.095	1187	.224	1400	.397	1580	.587
600	1060	580	.016	827	.053	994	.108	1245	.243	1452	.415	1632	.618
700	1237	677	.026	903	.067	1062	.123	1307	.265	1505	.440	1683	.643
800	1413	773	.038	982	.084	1133	.142	1370	.292	1562	.466	1737	.675
900	1590	870	.054	1062	.105	1207	.166	1440	.319	1623	.504	1792	.707
1000	1767	967	.075	1143	.130	1283	.194	1507	.350	1692	.543	1850	.753

Size 30

Velocity through Wheel	Capacity Cu. Ft. Air per Min.	0" S.P.		⅛" S.P.		¼" S.P.		½" S.P.		¾" S.P.		1" S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
500	2450	290	.026	452	.122	556	.264	712	.622	840	1.10	948	1.63
600	2940	348	.044	496	.148	596	.299	747	.676	871	1.15	979	1.72
700	3440	406	.071	542	.185	637	.341	784	.737	903	1.22	1010	1.79
800	3930	464	.106	589	.234	680	.395	822	.810	937	1.29	1042	1.88
900	4420	522	.151	637	.291	724	.460	864	.886	974	1.40	1075	1.96
1000	4910	580	.208	686	.360	770	.539	904	.972	1015	1.51	1110	2.09

Size 36

Velocity through Wheel	Capacity Cu. Ft. Air per Min.	0" S.P.		⅛" S.P.		¼" S.P.		½" S.P.		¾" S.P.		1" S.P.	
		R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.	R.P.M.	Hp.
500	3535	242	.037	377	.176	463	.380	593	.896	700	1.59	790	2.35
600	4241	290	.064	413	.213	497	.431	623	.973	726	1.66	816	2.47
700	4948	338	.103	452	.266	531	.491	653	1.06	753	1.76	842	2.57
800	5655	387	.153	491	.337	567	.569	685	1.17	781	1.86	868	2.70
900	6362	435	.217	531	.419	603	.662	720	1.28	812	2.02	896	2.83
1000	7069	483	.299	572	.518	642	.776	753	1.40	846	2.17	926	3.01

*The above table has been greatly reduced. Complete tables can be secured from the manufacturer.

A few of the more common fan laws are given as follows:

I. For a given fan size, piping system, and air density:
    (a) When speed varies
        (1) Capacity varies directly as the speed ratio.
        (2) Pressure varies as the square of the speed ratio.
        (3) Horsepower varies as the cube of the speed ratio.
    (b) When pressure varies
        (4) Capacity and speed vary as the square root of the pressure.
        (5) Horsepower varies as the (pressure)$^{3/2}$

II. For a constant pressure, density and point of rating:
  (a) When fan size varies
    (6) Capacity and horsepower vary as the square of the fan size.
    (7) Speed varies inversely as the fan size.
III. The above may be combined for one convenient operation.
  (a) When speed and fan size both vary
    (8) Capacity varies as ratio size^3 × ratio r.p.m.
    (9) Pressure varies as ratio size^2 × ratio r.p.m.^{2}
    (10) Horsepower varies as ratio size^5 × ratio r.p.m.^{3} or horsepower varies as ratio capacity × pressure.
IV. For constant pressure
  (a) When density varies
    (11) Speed, capacity, and horsepower vary inversely as the square root of the density, that is, inversely as the square root of the barometric pressure and directly as the square root of the absolute temperature.
V. For constant capacity and speed
  (a) When density of air varies
    (12) Horsepower and pressure vary directly as the air density, that is, directly as the barometric pressure and inversely as the absolute temperature.
VI. For constant amount by weight
  (a) When density of air varies
    (13) Capacity, speed, and pressure vary inversely as the density, that is, inversely as the barometric pressure and directly as the absolute temperature.
    (14) Horsepower varies inversely as the square of the density, that is, inversely as the square of the barometric pressure and directly as the square of the absolute temperature.
  (b) When both temperature and pressure vary
    (15) Capacity and speed vary as $\sqrt{\text{pressure} \times \text{absolute temp.}}$
         Horsepower varies as $\sqrt{\text{pressure}^3 \times \text{absolute temp.}}$

While discussing fan performance it might be well to consider the characteristics of the flow of air in ducts and the effects of a damper.

Let us assume a duct system that is calculated to pass 11,000 c.f.m. and requires $\frac{3}{4}$-inch static pressure to maintain this flow. A static pressure curve of duct resistance may be plotted through this point based on the fact that the pressure required to overcome resistance to flow varies substantially as the square of the capacity flowing, see curve A plotted on PD-8888 (Fig. 157).

Now suppose it is desired to use a Size 6 fan for supplying the air and that we wish it to be direct connected to a motor operating at 575 r.p.m. Such a fan pressure has been drawn and labeled curve D. It is evident that only one condition of static pressure and capacity satisfies both the fan and duct characteristics and it is at the intersection R of the two pressure curves, namely, 0.96″ s.p. and 12,400 c.f.m. Under these conditions it may be seen from the horsepower curve E that 3.08 hp. will be required.

If it is desired to obtain the exact air flow for which the duct system was laid out (11,000 c.f.m. at $\frac{3}{4}$″ s.p.) either of two methods may be employed; namely,
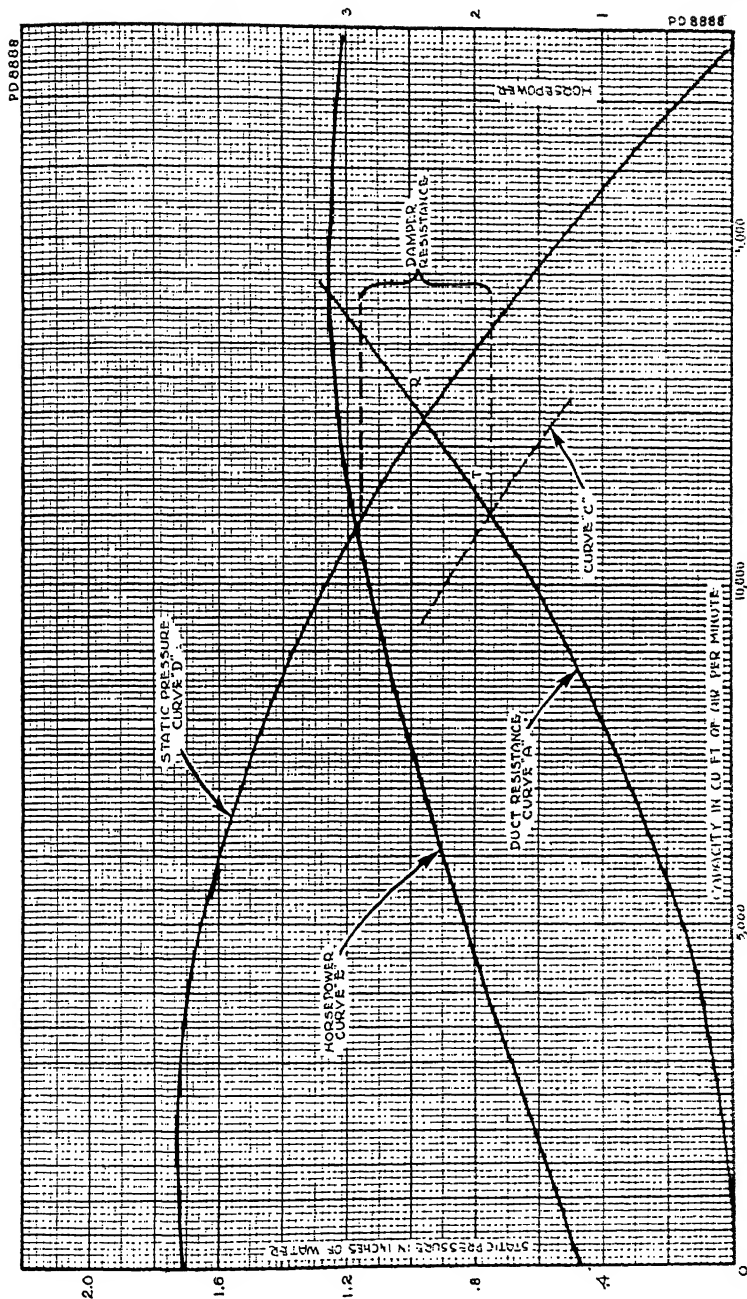
Fig. 157. Static Pressure

Courtesy of Buffalo Forge Company, Buffalo, N. Y.

by dampering or by speed change, the latter method requiring the lesser power but necessitating recourse to a belt drive or a variable speed motor.

Should we wish to use the damper method, it will be noted that at 11,000 c.f.m. and 570 r.p.m., the fan pressure is 1.16″ and therefore a damper resistance equivalent to .41″ at 11,000 c.f.m. will be required. Curve E shows that the necessary horsepower will be 2.95.

If, instead of dampering, we reduce the fan speed in the ratio of 11,000 ÷ 12,400 we find that the new fan curve C at 510 r.p.m. satisfies our condition. It should be observed that the point of rating R moves back along the duct resistance curve to T by the same rule as the frictional resistance of the duct itself (see Laws 1 and 2 preceding). Since the horsepower varies as the cube of the fan speed only 2.15 hp. will be required at 510 r.p.m. for the 11,000 c.f.m. at $\frac{3}{4}$″ s.p.

Let us assume several examples for the purpose of illustration.

Example 1. Select a quiet ventilating fan to handle 10,000 c.f.m. at 1″ s.p. Fan to be belt driven. Outlet velocity not to exceed 1,200 feet per minute. From the CL fan multirating tables (Table 68) select a No. 6½CL—outlet velocity 1,170 feet per minute.

Rating: 10,000 c.f.m. 1″ s.p. 70°F., 463 r.p.m. 2.19 hp.
The above rating is obtained by interpolation.

Example 2. If the fan size, piping system, and air density remain the same as in Example 1, but the fan is direct connected to a 570 r.p.m. motor, what is the rating?

Capacity	$= 10,000 \times 570/463$	$= 12,300$	(Law 1)
Static Pressure =	$1 \times (570/463)^2 =$	1.52″	(Law 2)
Hp.	$= 2.19 \times (570/463)^3 =$	4.09	(Law 3)

Example 3. If the fan size, piping system, and air density remain the same as in Example 2, but the system resistance is increased from 1.52″ to $1\frac{3}{4}$″ what is the rating?

$$\text{Capacity } 12,300 \times \sqrt{\frac{1.75}{1.52}} \qquad = 13,200 \qquad \text{(Law 4)}$$

$$\text{Speed } \quad 570 \times \sqrt{\frac{1.75}{1.52}} \qquad = \quad 611 \qquad \text{(Law 4)}$$

$$\text{Hp. } \qquad 4.09 \times \sqrt{\left(\frac{1.75}{1.52}\right)^3} \quad = \quad 5.05 \qquad \text{(Law 5)}$$

Example 4. Given the No. 6½ CL of Example 1 whose rating is: 10,000 c.f.m. 1″ s.p. 70°F., 463 r.p.m. 2.19 hp. Find the capacity, speed, and horsepower of a No. 10 CL at the same pressure (1″)

Capacity	$= 10,000 \times (10/6\frac{1}{2})^2 = 23,600$		(Law 6)
Speed	$= 463 \times 10/6\frac{1}{2} = 712$		(Law 7)
Hp.	$= 2.19 \times (10/6\frac{1}{2})^2 = 5.19$		(Law 6)

Example 5. Given the No. 6½ CL of Example 1 at 463 r.p.m. Find the same point of rating of a No. 10 CL @ 570 r.p.m. (i.e., the point at which the efficiency of the No. 10 is the same as that of the No. 6½).

Capacity	$= 10,000 \times (10/6\frac{1}{2})^3 \times (570/463) = 44,700$	(Law 8)
Static Pressure =	$1 \times (10/6\frac{1}{2})^2 \times (570/463)^2 = 3.59''$	(Law 9)
Hp.	$= 2.19 \times (10/6\frac{1}{2})^5 \times (570/463)^3 = 35.15$	(Law 10)

Example 6. A case frequently met in selecting fans is one for which the air to be handled is specified at some temperature other than the standard of the fan tables (70°F.—29.92″ bar.). Usually the volume so stated will be measured at the temperature given, but occasionally in the case where the fan is drawing through a heater, the volume stated may be standard conditions. However, in any case the conditions should be clearly stated.

It should be borne in mind that at any definite speed a fan will handle a definite cubic quantity regardless of the density.

Select a fan to handle 10,000 c.f.m. at $\frac{1}{2}''$ s.p. and 335°F. Equivalent pressure at 70° = $\frac{1}{2} \times (795/530) = \frac{3}{4}''$. From multirating Buffalo Forge Company type CL fan tables select a No. 6 CL. (Table 68.)

Rating: 10,000 c.f.m. $\frac{1}{2}''$ s.p. @ 335°F., 481 r.p.m. 1.19 hp. @ 335°F.
1.78 hp. @ 70°F.

The limit load table shows that at 481 r.p.m. the maximum power which the fan can take at 70°F. is 1.82 hp. thus assuring that a 2 hp. motor will never be overloaded, even with a 10% belt loss.

Example 7. If the fan in the preceding example is to draw 10,000 c.f.m. of 70°F. through a heater, which heats the air to 335°F. and the resistance of the system is $\frac{1}{2}''$ when handling 10,000 c.f.m. of 335°F. air, what is the rating of the fan? Capacity is proportional to the absolute temperature.

Capacity increase	$= 10,000 \times$	$795/530$	$= 15,000$ c.f.m.
Speed	$=$	$481 \times 15,000/10,000 =$	771 r.p.m.
Static Pressure	$=$	$\frac{1}{2} \times (771/481)^2$	$= 1.12''$ @ 335°
Hp.	$=$	$1.19 \times (771/481)^3$	$= 4.01$ hp. @ 335°

Example 8. Select a fan to handle 10,000 c.f.m. of 140°F. air at an altitude of 1600′ against a static pressure of $1\frac{1}{4}''$ measured at the temperature and altitude. By Law 12 the equivalent pressure at 70° and sea level is

$$1\frac{1}{4} \times \frac{460+140}{460+70} \times \frac{29.92}{28.15} = 1\frac{1}{2}''$$

From the CL fan multirating tables select a No. 6 CL 1375 outlet velocity.

10,000 c.f.m. $1\frac{1}{4}$ Pressure @ 140° and 1600′
$1\frac{1}{2}$ Pressure @ 70° and S.L. 609 r.p.m. 3.26 hp. @ 70° S.L.
3.06 hp. @ 70° 1600′
2.72 hp. @ 140° 1600′

In all the preceding examples the static pressure or the loss through the system was given. The following example shows the calculation of the losses of a typical exhaust system.

In Fig. 158 the system is based on a velocity of 4,500 feet per minute in the branch pipes with the area of the main pipe equal to the area of the preceding branch pipes plus 20%. It is necessary to figure only the friction in the greatest run of pipe. The air flow in the other branches is taken care of by dampers.

Beginning with the 6-inch pipe we find that there are 13 feet or 26 diameters. There are also two right angle turns which are roughly equivalent to 10-pipe diameters each. This is a total of 46 diameters. We may assume one head loss for every 55 diameters of pipe. Therefore, we have $46/55 = .84$ head loss. There
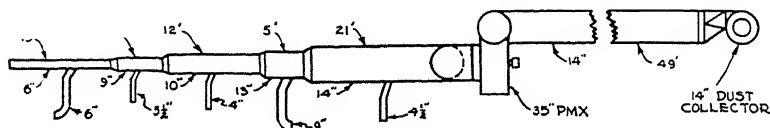
Fig. 158. Dust Collector System

is also $\frac{1}{2}$ head loss at the entrance plus the one head required to create a velocity in the pipe. Therefore, the total static required in the first section is $.84 + 1.5 =$ 2.34 heads. One head at 4,500 vel. $= \left(\dfrac{4,500}{4,000}\right)^2 = 1.26''$ water, 2.34 heads $= 2.95''$ water. The same method has been pursued for each size of main pipe and results tabulated below. One head is added for loss through collector.

Table 70. Pipe Data

Diam. Ins.	Length Ft.	Length Diams.	Add for Elbows	Total Diams.	Head Loss	Vel. in Main	Vel. Hds. Main	Static Loss
6	13	26	20	46	.84+1.5	4,500	1.26	2.95
9	6	8	..	8	.14	3,680	.85	.12
10	12	14.4	..	14.4	.26	3,700	.86	.22
13	5	4.6	..	4.6	.09	3,820	.91	.08
14	21	18	20	38	.69	3,760	.88	.61
14	49	42	10	52	.95+1	3,760	.88	1.72

Strictly speaking, there will be a slight loss at each change of section due to velocity changes but it is small enough to be negligible. Adding all the losses gives $5.7''$ as the pressure required at the fan.

From the PMX fans (Table 64) select a size 35 standard mill exhauster. Since the velocity in the main is 3,760 feet per minute we will need 4,020 c.f.m. The nearest rating is 4,268 c.f.m. 6" s.p. and 1,546 r.p.m. If the fan is run at 1,500 r.p.m. it will meet the needs of the system.

The following velocities, which should be used for conveying, are from the Industrial Code, Department of Labor, State of New York, Jan. 1, 1931.

Light Shavings..........2,400*	Cotton................4,000–5,000
Dry Sawdust...........3,000*	Rags...................4,000–5,000
Knots, Blocks..........3,600–4,000*	Dried Pulp............4,500–5,000
Wheat................5,500–6,000	Salt.................5,500–6,500
Corn.................5,500–6,000	Sand................6,500–7,500
Oats.................4,500–5,000	Powdered Coal.........4,000
Wool.................4,500–5,500	

Other types of fans or blowers can be selected and calculated by reference to the manufacturers' catalogues. The tables given here are only to show how such tables are used.

*Exhaust systems only, use higher velocities for conveying.

FIRING EQUIPMENT

Gas Burners. Generally speaking, gas burners are of two classes —(1) those used in furnaces or boilers designed for gas fuel and (2) those used in furnaces or boilers originally designed for coal fuel. Burners in the first class are designed as a part of the heating equipment and their selection should be governed by the ratings for the furnace or boiler as a whole. The burners in the second class are known as "conversion burners" because they are designed to convert coal-fuel heaters into gas-burning heaters.

Conversion burners are often designed so that radiants or refractories (see Figs. 160 and 163) are used to convert some of the energy in the gas to radiant heat. Because of the low heat transfer that takes place in flue passages, designers have tried to transfer most of the heat from the gas to these mediums to be heated within the firepot. Other burners are of the blast type with luminous flames operating without refractories.

Some conversion burners have sheet metal air ducts which are inserted through the ashpit door. (See Fig. 159.) The duct has automatic air controls which remain open when the burner is operating and closed when the gas is shut off. This prevents circulation of cold air through the burner space when it is not in operation, and only the air needed for combustion is supplied. Conversion burners are manufactured in various shapes to be used with different furnaces. Types vary from the fully automatic to varying automatic and manual control. Gas is especially applicable to automatic control.

A gas-fired boiler under thermostatic control is so sensitive to variations in temperature that generally a sufficient pick-up load is only 25% greater than the equivalent standard cast-iron column radiation which the boiler serves. Manually controlled gas burners may require as great as 100% pick-up load to overcome the starting load and losses in piping.

Table 71 gives selection factors to be added to the installed steam radiation under thermostatic control. These same factors may

be used in determining the gas demands for which conversion burners, installed in steam or hot water boilers, should be set. The proper hourly rate of gas consumption may be determined by multiplying the equivalent direct heating surface (radiation) by 240, adding the appropriate percentage from Table 71, dividing by the heat value of the gas and again by the heating efficiency.

Table 71. *Selection Factors for Gas Boilers

Cast-Iron Steam Radiation (Equivalent Square Feet)	Selection Factor (Per Cent)	Cast-Iron Steam Radiation (Equivalent Square Feet)	Selection Factor (Per Cent)
500	56.0	2,000	45.0
800	54.0	3,000	42.5
1,200	51.0	4,000 and over	40.0
1,600	48.0	.....	....

*Courtesy of A.S.H.V.E. Guide, 1936, Chapter XXVIII.

GAS WARM AIR FURNACES†
Construction Requirements

Section 1. *Burners.* (a) The burner head shall be a single casting or shall be of an equally gas tight and durable construction.

(b) Ports shall be machined or otherwise accurately made.

(c) Burner assemblies shall not depend upon cement joints for tightness.

(d) Gauzes shall not be used in burner construction.

(e) Burners shall be easily removable.

(f) Burners shall be so secured that they will not twist, slide or drop out of position.

(g) Provision shall be made to permit ready observation of all flames under operating conditions.

Section 2. *Air Mixers.* (a) Air mixer faces shall be smoothly finished.

(b) Provision shall be made to maintain, under reasonable conditions of usage, the relative positions of air mixers, mixing tubes, and orifices.

Section 3. *Air Shutters.* (a) Air shutters of atmospheric burners shall make a close fit with the mixer face.

(b) Air shutters shall be capable of any desired adjustment.

(c) Air shutters shall be of substantial and durable construction and so made that they can be securely fixed in any desired position.

Section 4. *Pilots.* (a) Every furnace shall be equipped with a pilot or equivalent means for igniting the gas at the main burner or burners.

(b) Pilot burners shall be supported in such a manner that their position relative to the main burners is fixed.

(c) Pilots shall be so placed that they can be easily seen, safely lighted, and readily removed.

†Approval Requirements for Central Heating Gas Appliances, from Standard Published by American Gas Association, Sixth Edition.

(d) Bunsen type pilots shall be so constructed that ignition of the main burner flames occurs in a normal manner, even though the pilot flame is burning at the orifice. A pilot burner that cannot be made to flash back under any conditions of test shall be interpreted as meeting the requirements of this clause.

(e) Where the enclosed type of fire box is used, special provision shall be made to insure an adequate supply of air for combustion of gas from pilot.

(f) Where iron pipe is used for pilot burner manifolds, its nominal size shall be not less than $\frac{1}{4}$ inch and where used for individual pilot lines its nominal size shall be not less than $\frac{1}{8}$ inch. It shall comply with the American Tentative Standard for Wrought-Iron and Wrought-Steel pipe, A. S. A. B36.10-1935.

(g) Where metallic tubing is employed its inside diameter and wall thickness shall be not less than 0.125 and 0.030 inch respectively and it shall comply with the applicable American Standard Listing Requirements for Semi-Rigid Gas Appliance Tubing and Fittings, Z21.24-1936.

(h) When the pilot burner supply line is taken from a horizontal line, the connection shall be made either on the side or top. If taken from a vertical line, it shall be above the main burner supply line.

(i) A fixed orifice shall be provided independent of the operation of the pilot gas valve for limiting the amount of gas consumed by the pilots. The maximum normal rate obtainable by such means shall neither exceed 5 cubic feet per pilot per hour at 7 inches water column pressure, nor shall the total gas used by all the pilots in a furnace exceed 10% of the rated input, except that a pilot rate of less than 2 cubic feet per hour per pilot shall not be required.

Where a separate gas pressure regulator is installed in the pilot line, the adjustment outlet pressure specified by the manufacturer shall be employed in determining the maximum rating of the pilot in accordance with this requirement.

Section 5. *Main Burner Valves.* (a) Main burner valves shall be provided to control the gas to each separate main burner having an input in excess of 30,000 B.t.u. per hour except as noted:

(1) Where the gas supply to the main burner or burners is controlled separately from the pilot gas supply by the main control valve of the appliance and where the furnace is provided with a single combustion chamber and equipped with less than three main burners and the main control valve specified in Section 6 is in such location that it is accessible from the position assumed in lighting the pilot or main burner, no burner valves shall be required.

(2) Where the gas supply to the main burner or burners is controlled separately from the pilot gas supply by the main control valve of the appliance and where automatic devices to prevent the escape of unburned gas are provided for each individual burner, no burner valves shall be required.

(3) No main burner valves shall be permitted on furnaces equipped with power burners.

(b) Burner valves of the plug type shall meet the minimum dimensions shown for circumferential seal, spring take-up and seal above and below the gasway.

(c) Main burner valves shall have stops at the full **on** and full **off** positions.

(d) Burner valve handles shall be made of metal and so constructed that they will indicate at a glance the **on** and **off** positions.

(e) Valve handles shall be cast integrally with or otherwise permanently attached to plugs.

(f) Main burner valves shall be so constructed that they can be turned on or off easily by hand.

Section 6. *Main Control Valve.* (a) A properly labeled, manually operated valve shall be provided to control the gas supply to the main burner manifold.

(b) Where all main burners are not equipped with individual valves, the main control valve shall be located downstream from the pilot connection and the pilot line shall be equipped with an individual control valve.

Section 7. *Automatic Devices to Prevent Escape of Unburned Gas.* (a) Every furnace, except floor furnaces, shall be equipped with an automatic device which will prevent the escape of unburned gas from the main burner or burners.

(b) Every automatically or remotely controlled floor furnace shall be equipped with an automatic device which will prevent the escape of unburned gas from the main burner or burners.

(c) These devices supplied on gas furnaces shall comply with the applicable construction requirements specified in the American Standard Listing Requirements for Automatic Devices Designed to Prevent Escape of Unburned Gas, Z21.20-1935.

Section 8. *Gas Pressure Regulators.* (a) A gas pressure regulator which will satisfactorily limit the gas supply pressure shall be supplied with each gas furnace, including floor furnaces.

(b) Gas pressure regulators supplied on gas furnaces shall comply with the applicable construction requirements specified in the American Standard Listing Requirements for Domestic Gas Appliance Pressure Regulators, Z21.18-1934.

Section 9. *Electric Gas-Control and Diaphragm Valves.* (a) Electric gas-control and diaphragm valves supplied on gas furnaces shall comply with the applicable construction requirements specified in the American Standard Listing Requirements for Automatic Main Gas-Control Valves, Z21.21-1935.

(b) Where an electric valve is used to control the gas supply to main power burners, no means for manual operation shall be permitted.

(c) Electric valves shall bear a label which shall indicate type of current, wattage, and in the case of alternating current, the frequency.

(d) Electric valves designed for two or more rates shall be equipped with readily accessible leak-proof adjustment screws for regulating all rates other than the full open rate, except in the case of power burners where the valve shall be so constructed that no adjustment of the rate of flow of the gas is permitted, unless the burner system is such that adjustment of the valve will not affect the proportion of air and gas supplied to the burner.

*Section 10. *Electric Ignition.* (a) If electric ignition is used, the means for igniting the gas shall be so designed and located as to eliminate the collection of carbon or other materials, or the dislocation, distortion, or burning of parts as the result of normal conditions of heating or vibration of parts.

(b) Every automatically or remotely controlled system employing electric ignition shall be so designed as positively to prevent delayed ignition, reduce ignition failure to a minimum, and shut off the gas supply in the event of failure to cause ignition. Such systems shall meet the applicable performance requirements specified herein for pilots and for ignition.

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*This section and Sections 11, 12, 13 and 14 which follow have been adopted from Subject 296, Underwriters Laboratories Standards for Construction and Performance of Fuel Oil Burners for Domestic Use, dated March, 1934, paragraphs 87 to 101, inclusive.

Section 11. *Electric Ignition Transformers.* (a) High tension terminals for transformer and wire leads shall be designed to provide protection from shocks.

(b) High tension terminals on transformers shall be so located on the appliance as to be free from contact with metal parts, such as adjustable legs, valves, or controlling mechanism base-plates, or housings.

(c) Transformers shall be placed in such a position as to be free from temperatures likely to destroy the insulation used therein.

Section 12. *Electric Ignition High Tension Leads.* (a High tension leads shall consist of the usual type of gas-tube sign and oil-burner ignition cable complying with the Underwriters Laboratories Standard for Gas-Tube Sign and Oil-Burner Ignition Cable, First Edition, January 1936, Subject S14. Such leads shall be provided at each end with brass loops, eyes, or other equivalent means to facilitate and insure rigid connections to binding posts.

(b) High tension leads or cables shall be as short as possible, free from any sharp bends, and be protected from rough usage, soakage with water or steam or abrasion.

Section 13. *Electric Ignition Electrodes and Bus Bars.* (a) Electrodes and bus bars of the non-insulated type shall be suspended away from all metal parts and be so insulated and arranged as to be free from arcovers at any point throughout the assembly when a voltage 50 per cent in excess of the maximum possible voltage to ground is impressed for a period of one minute between the normal point of transformer connection and ground. Such tests shall be conducted both before and after the normal furnace tests.

(b) Electrodes or bus bars supporting electrodes shall be so designed that they may be readily locked into proper position and no adjustment of any mechanical nature shall be allowed in electrodes of this type.

(c) Electrode tips shall preferably be of the pointed type or of the knife-edge type and so designed that extreme burning of the points will not result over a period of time.

(d) Flexibility in electrodes at their outer ends may be permitted if designed to resist warping and accidental dislocation.

Section 14. *Electric Ignition Insulators.* (a) Insulators shall consist of high-grade porcelain or equivalent non-combustible insulating material. Such insulation material shall be glazed or otherwise made impervious to internal collection of moisture from the gas and shall be readily cleanable.

(b) Insulators shall not be used where accumulations of carbon may form.

Section 15. *Fan Furnace Limit Control.* Furnaces, except floor furnaces, which include fans as integral parts of their construction, shall be provided with means which will shut off the gas supply to the main burners when the temperature in the bonnet reaches 200°F., and, if adjustable, such means shall be provided with a stop which will not permit an indicated adjustment of more than 250°F.

Section 16. *Gas Piping and Controls.* (a) Gas piping and or gas controls shall not be located within circulating air passages.

(b) Where semi-rigid tubing is employed its inside diameter and wall thickness shall be not less than 0.125 and 0.030 inch respectively, and it shall comply with the applicable requirements specified in the American Standard Listing Requirements for Semi-Rigid Gas Appliance Tubing and Fittings, Z21.24-1936.

Section 17. *Draft Hoods.* (a) A suitably designed draft hood shall be supplied with each furnace.

(b) A draft hood shall be made of such material that it will not become distorted when the furnace is operated at 1.5 normal gas pressure with the hood supporting a load equal to 10 pounds per inch of nominal outlet diameter.

Method of Test

The draft hood shall be connected to the furnace in accordance with the manufacturers' instructions. A vertical compression load equal to that specified above shall be applied without impact to the outlet of the hood in such a manner that no obstruction will be offered to the ready escape of the flue gases.

In the case of a draft hood with a horizontal flue outlet, a 90-degree sheet metal elbow with inlet and outlet connections of the same nominal size as the outlet collar of the hood, with no abrupt bends, and with reasonably smooth inner contour shall be attached to the flue outlet of the hood. A load equal to that specified shall then be applied without impact to the outlet of the elbow.

The gas rate shall be adjusted at normal pressure to within plus or minus 2% of the manufacturer's hourly B.t.u. input rating and the gas ignited. The gas pressure shall then be increased to 1.5 normal pressure. The appliance shall then be operated for at least 15 minutes after equilibrium flue temperature conditions have been reached.

If at the end of the period specified no distortion of any part is noted, the provisions of this requirement shall be deemed met.

(c) The construction of a floor furnace shall be such that a clearance of at least 15 inches is provided between the bottom of the floor grille and the top of the outlet of a draft hood having a horizontal flue outlet and the top of a 90-degree sheet metal elbow attached to the outlet of a draft hood having a vertical flue outlet.

(d) All parts of the draft hood shall be constructed of a material having a thickness not less than that of No. 26 U. S. Standard gauge sheet metal.

(e) No flue damper shall be used.

(f) The outlet collar of a draft hood shall be of such size as to accommodate flue pipe of integral inch diameter.

(g) Detachable draft hoods shall be so designed that removal and replacement in normal usage will not permanently deform any part nor alter the relative position of any part with respect to another.

Section 18. *Flue Connections.* Flue connections shall extend beyond the furnace casing a sufficient distance to insure secure fastening of the flue pipe.

Section 19. *Primary and Secondary Air Intake.* All primary and secondary air for burners must be drawn from the outside of the furnace casing or from furnace openings which extend through the outside casing.

Section 20. *Furnace Openings.* Furnace openings other than flue connections, such as secondary air supplies, doors, etc., shall extend through the outside casing a sufficient distance to permit secure fastening of outside frames and to prevent possible leakage of combustion products into the warm air ducts.

Section 21. *Disposal of Combustion Products.* The construction of the furnace shall be such that no part of the products of combustion becomes mixed with the warm air discharged by the furnace.

Section 22. *Joints in Heating Surfaces.* All joints of heating surfaces shall be of welded, brazed, screwed, lock seamed, machined and bolted, or of formed,

slip joint, or flanged construction, tightly bolted together and enclosing gaskets or other similar packing materials.

Section 23. *Thickness of Material.* (a) Where outside casings are constructed of sheet iron or steel, they shall be not less than No. 26 U. S. Standard gauge for furnaces of less than 100,000 B.t.u. per hour input capacity. For furnaces with a capacity in excess of this amount, the thickness of the furnace casing shall be not less than No. 24 U. S. Standard gauge. Where reinforcing or a double construction is employed, strength equivalent to that afforded by weights of metals mentioned above shall be regarded as acceptable.

(b) Where sheet metal is used in the construction of heating surfaces, the thickness shall be such as to insure strength, rigidity, durability, resistance to corrosion, and other physical properties equivalent to No. 22 U. S. Standard gauge black sheet iron.

Section 24. *Accessibility for Cleaning.* The flue gas passage-way through furnaces shall be accessible for cleaning where:

(1) The products of combustion are drawn below the level of the burner.

(2) The temperature of the combustion products is less than 250°F. when the furnace is operated at normal pressure with the gas rate within plus or minus 2% of the manufacturer's hourly B.t.u. input rating.

(3) The width of any flue gas passage is less than $1\frac{1}{2}$ inches.

Section 25. *Air Circulation in Floor Furnaces.* (a) Floor furnaces shall be so constructed and designed that they will provide a constant circulation of heated air at all times during their operation.

(b) Floor furnaces with a single warm air register located vertically or at an angle shall not be equipped with a damper designed to restrict the flow of warm air from the furnace.

(c) Floor furnaces having two warm air registers located either vertically or at an angle when equipped with a damper shall be so designed that only one or a fraction of one opening may be closed simultaneously with the damper.

Section 26. *Observation Doors.* Observation doors of floor furnaces shall fit tightly and shall be so machined or of such weight and construction that they will not be displaced by any normal jar or pressure created during normal operation of the furnace.

Section 27. *Assembly.* (a) The construction of every part of the furnace shall be such that it will not show signs of becoming so warped, bent, or broken as to prevent its compliance with any of these requirements.

(b) Construction not covered by these requirements shall be in accordance with reasonable concepts of safety, substantiality, and durability.

Section 28. *Name Plate.* All gas furnaces shall bear a permanent name plate on which shall appear in permanent form:

(1) The manufacturer's or distributor's name.

(2) The manufacturer's or distributor's number of the appliance.

(3) The manufacturer's hourly B.t.u. input rating.

(4) The output rating in B.t.u. per hour.*

Figs. 159 and 160 show an external and an internal view of a typical conversion gas burner. Fig. 161 shows an enlarged view of the

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*This shall be computed as follows: For furnaces: hourly B.t.u. input rating × 0.75 = output rating. For floor furnaces: hourly B.t.u. input rating × 0.70 = output rating.

control unit. Table 72 gives ratings for typical conversion burners.

The conversion burner in Fig. 159, for example, presents a compact, streamlined appearance. The heavy-gauge sheet metal duct is finished in crackle finish with all metal parts chromium plated. All controls, including the gas pressure regulator, are completely enclosed in the duct. The rounded chromium louvers admit air to the control
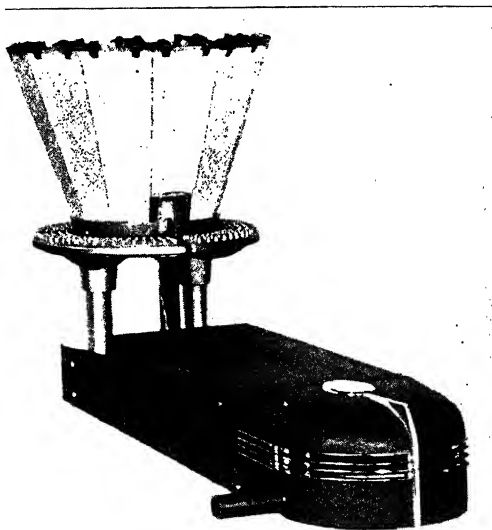
Fig. 159. Conversion Gas Burner
Courtesy of Bryant Heater Company, Cleveland, Ohio

chamber. From the control chamber the total air supply to the burner passes to the burner duct through an automatic air door located in a rigid cast-iron frame.

Table 72. *Type R Round Burner

Burner Number	Rating B.t.u. Per Hour Input		Min. Fire Box Diam. Inches	Control Size Inches	No. of Heads and Tubes	Weight in Pounds
	Max.	Min.				
2-R-95	150,000	60,000	15	1	1	110
3-R-95	200,000	90,000	17	1	2	125
4-R-95	300,000	125,000	21	1	2	155

*Courtesy Bryant Heater Company.

It is necessary merely to connect a 1-inch gas supply pipe to the concealed control valve as shown in Fig. 160. There are no projecting handles or adjustments to be damaged in shipment or tampered with by children. The solenoid valve is mounted in the control chamber with the manual control knob extending level with the top of the control cover. The pilot lighter button is mounted $\frac{1}{4}$ inch below the top of this cover with a hole provided for operation of the button.
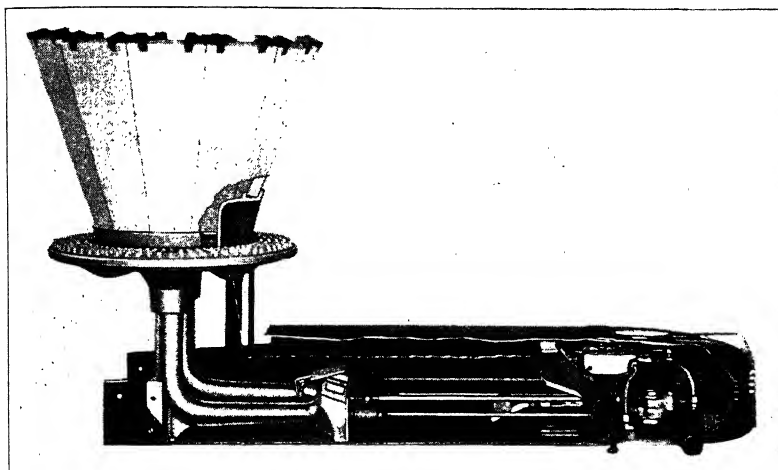
Fig. 160. Cutaway View of Conversion Gas Burner
Courtesy of Bryant Heater Company, Cleveland, Ohio

The control chamber cover can be removed, providing complete access to all burner adjustments and controls. This front portion of the duct serves as a protecting cover, enclosing the adjustments and controls, and as a means of beautifying the external appearance of the burner. Its removal in no way affects the operation or adjustment of the burner. All burner adjustments can be made with the control cover removed.

The burner tubes are bolted in position in the burner duct. To use the burner in a furnace, it is necessary only to insert the assembly in the ashpit door, bolt the pilot in an upright position, and mount the burner heads, baffle bowl, and baffles in place.

Two adjustable front legs are provided for setting the burner at the proper height. The height of the duct is 6 inches so the duct

assembly can be raised without interfering with the top of the ashpit opening. The width of the duct is 11 inches.

The length of the duct is designed so no special extensions will be necessary on warm air installations. The ashpit closing surface can enclose the duct at any point up to the control chamber cover. There are no exposed air door or control adjustments which prevent the insertion of the burner duct into the ashpit up to this joint.
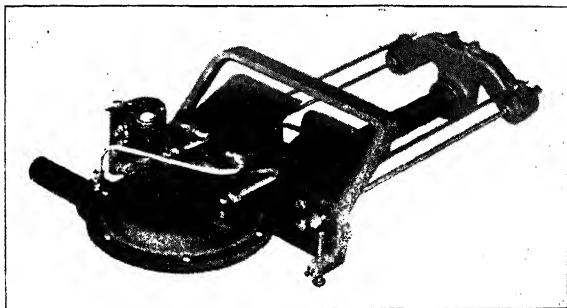
Fig. 161.   Control Unit for Conversion Gas Burner
Courtesy of Bryant Heater Company, Cleveland, Ohio

A separate air door operating diaphragm is connected directly to the top of the snap valve diaphragm so the same gas which closes the snap valve also closes the air door. The door opens by gravity. An automatic flue damper can be connected in at this point.

A properly designed conversion burner baffle should perform two essential functions: (1) It should serve as a means of converting a portion of the heat into radiant heat to be directed *horizontally outward* against the fire box walls. (2) It should serve as a means of obtaining a scrubbing action of the hot combustion gases over the fire box walls. Projections on the radiant surface become incandescent on the lower surface only, directing radiant heat downward upon the burner heads and into the ashpit. This same fact is true of baffles designed to be leaned against the fire box wall—especially when the angle of the baffle is increased.

The thermostatic pilot assembly for burners, such as Fig. 159, is enclosed in a 1-inch square rigid steel housing. The pilot stem is of stainless steel with a slotted carry-over port. It is practically impossible to blow out the pilot because the pilot flame extends from three sides of the stem and the pilot is protected by a cast-iron housing

which is part of the baffle bowl (round burner). The pilot gas supply connection is made on the inlet side of the gas pressure regulator—a feature which has become a requirement of a great many utilities. Under extreme line pressure reduction, the pilot flame will shorten sufficiently to allow the thermo-element to cool and the snap valve will close at a point where there is still sufficient pilot flame for safe burner ignition.

The thermo-element consists of two chrome-steel channels mounted back to back and pinned at three points in an A-frame. Surrounding temperature affects each channel equally so that no movement is transmitted. However, when one channel is heated by the pilot flame, it elongates and causes the A-frame to swivel. This movement makes and breaks the pilot contacts in the solenoid valve circuit. There is no deflection or bending of the channel elements. The movement is all straight-line elongation or contraction.

Extremely rapid pilot timing can be secured. The pilot timing adjustment screw is located in the front control chamber. As the thermo-element is compensated for surrounding temperatures very little change in timing is apparent in hot and cold fire boxes.

The pilot is definitely located in position so both burner heads will be instantly ignited. The pilot cannot be mounted in any position but the proper one. The entire pilot assembly can be drawn out through the duct for inspection and cleaning without disturbing the pilot adjustment. The removal of the entire pilot, however, will probably never be necessary, as both the pilot supply tube, which contains the pilot orifice, and the bleeder tube, which has no orifice, can be withdrawn from the pilot assembly without removing the pilot.

One feature of the burner is the safety of the outside pilot lighter. This lighter consists of a stationary heater coil adjacent to the pilot and a push button accessible through an opening in the control chamber cover. To light the pilot it is necessary merely to open the main valve in the gas supply line to allow gas to flow to the thermo-pilot, then to depress the push button. No auxiliary flare pilot is required. When cold unlighted gas issues from the pilot ports, it passes horizontally from the ports directly over the heater coil. However, as soon as the flame is ignited, the heat in the flame causes the jet to curl upward so the coil is not in the path of the actual pilot flame.

GAS INLET - 1" PIPE

$1\frac{1}{16}$

6

11

C DIA.

D DIA.

B

E

$12\frac{1}{16}$

A

$5\frac{3}{16}$

$4\frac{3}{4}''$ MAX. ADJUSTMENT
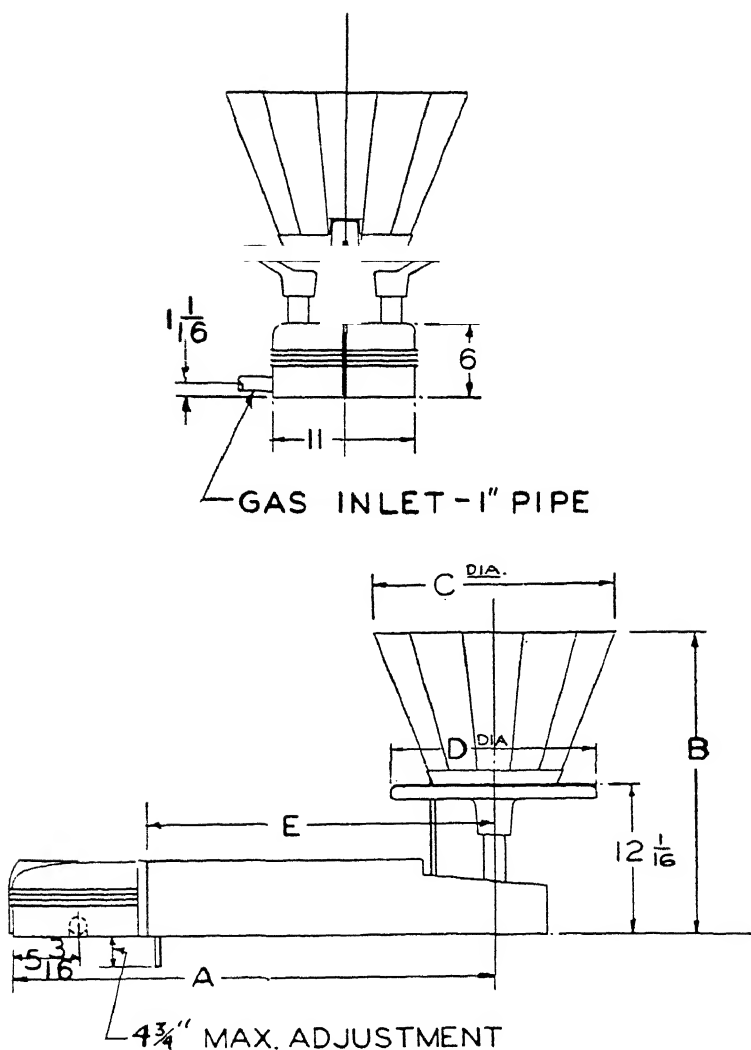
Fig. 162.   Typical Dimensions for Conversion Burners

Typical Dimensions for the Above Burner

MODEL	A	B	Max.	D	E
2-R-95	$37\frac{1}{16}$	$24\frac{1}{2}$	$16\frac{1}{2}$	14	$26\frac{3}{8}$
3-R-95	$38\frac{1}{16}$	$24\frac{3}{8}$	$18\frac{3}{4}$	16	$27\frac{3}{8}$
4-R-95	$40\frac{1}{16}$	$23\frac{3}{8}$	25	20	$29\frac{3}{8}$

Courtesy of Bryant Heater Company, Cleveland, Ohio

Fig. 162 shows dimensions for typical burners. Such dimensions are used in determining clearances necessary, for example, for installation of the burner in a warm air furnace.

There are many styles and makes of gas burners but the preceding discussion will serve to illustrate a typical burner and its ratings, operation, construction, and dimensions. Fig. 163 shows a rectangular conversion gas burner.
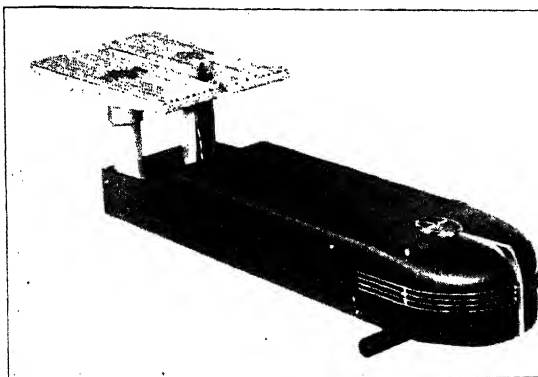
Fig. 163. Rectangular Conversion Gas Burner
Courtesy of Bryant Heater Company, Cleveland, Ohio

*Glossary of Terms. *Air Shutter.* An adjustable device for varying the size of the primary air inlet or inlets.

Appliances—Automatically Controlled. Appliances equipped with automatic devices which: (1) accomplish complete "turn-on" or "shut-off" of the gas to the main burner or burners; or (2) graduate the gas supply to the burner or burners but do not effect complete shut-off of the gas.

Appliance Flue. (See Flue.)

Atmospheric Burner. (See Burner.)

Baffle. An object placed in an appliance to change the direction of, or retard the flow of air, gas, air-gas mixtures, or flue gases.

Base. The lowest supporting frame or structure of the appliance, exclusive of legs.

B.t.u. Abbreviation for British thermal unit. The quantity of heat required to raise the temperature of one pound of water 1°F.

Burner. A device for the final conveyance of the gas, or a mixture of gas and air, to the combustion zone.

(1) Injection Burner. A burner employing the energy of a jet of gas to inject air for combustion into the burner and mix it with the gas at line pressure.

(a) Atmospheric Injection Burner. A burner in which the air at atmospheric pressure is injected into the burner by a jet of gas.

(2) Yellow Flame Burner. A burner in which secondary air only is depended on for the combustion of the gas.

(3) Power Burner. A burner in which either gas or air or both are supplied at pressures exceeding, for gas, the line pressure, and for air, atmospheric pressure; this added pressure being applied at the burner.

(a) Pre-Mixing Burner. A power burner in which all or nearly all of the air for combustion is mixed with the gas as primary air.

(4) Pressure Burner. A burner which is supplied with an air-gas mixture under pressure (usually from 0.5 to 14 inches of water and occasionally higher).

Burner Head. That portion of a burner beyond the outlet end of the mixer tube which contains the ports.

Burner Valve, Gas. A manually or mechanically operated valve which permits control of the flow of gas.

Central Heating Gas Appliance. A gas appliance, normally installed in the basement, for heating occupied rooms. Ordinarily this includes gas boilers, warm air furnaces, and floor furnaces, but does not include unit heaters, space heaters, nor industrial gas boilers.

Chimney Flue. (See Flue.)

Cock, Gas. A gas burner valve of the plug and barrel type.

Combustion. Combustion, as used herein, refers to the rapid oxidation of fuel gases accompanied by the production of heat or heat and light.

Combustion Chamber. The portion of an appliance within which combustion occurs.

Combustion Products. Constituents resulting from the combustion of a fuel gas with the oxygen of the air, including the inerts, but excluding excess air.

Condensate. The liquid which separates from a gas (including flue gases) due to a reduction in temperature.

Controls. Devices designed to regulate the gas, air, water and/or electrical supplies to a gas appliance. These may be manual, semi-automatic or automatic.

Cubic Foot of Gas. The amount of gas which would occupy one cubic foot when at a temperature of 60°F. if saturated with water vapor and under a pressure equivalent to that of 30 inches of mercury.

Diaphragm Valve. A device consisting essentially of a gas valve actuated by means of the application of gas pressure upon a flexible diaphragm.

Dilution Flue. (See Flue.)

Draft Hood. A device placed in, and made a part of the flue pipe from an appliance, or in the appliance itself, which is designed to (1) insure the ready escape of the products of combustion in the event of no draft, back-draft, or stoppage beyond the draft hood; (2) prevent a back-draft from entering the appliance; and (3) neutralize the effect of stack action of the chimney flue.

Drip. The container placed at a low point in a system of piping to collect liquid condensate and from which the condensate may be removed.

Drop. Any vertical pipe or nipple which conducts the gas down.

Electric Gas-Control Valve. A device actuated by electrical energy for controlling the gas supply.

(1) Motor Valve. An electric gas-control valve in which the valve is completely opened by the rotation of an electric motor and generally automatically

closed by a spring or other mechanical means in the event the electrical circuit is broken.

(2) Modulating Valve. A motor valve so designed that the valve opening is controlled within narrow limits throughout the entire range from the "full-open" to the "closed" position by means of the motor.

(3) Solenoid Valve. A valve which is opened or closed by the action of an electrically excited coiled wire magnet upon a steel bar attached to valve disc.

(4) Step (Manual) Valve. A valve having a rotating plug with, generally, three **on** positions and different rates of gas flow for each, the plug being actuated by a solenoid or motor-driven rack and pinion and a cam arrangement which together with a combination push-button switch determine the position assumed by the plug.

Excess Air. Air which passes through the combustion chamber and the appliance flues in excess of that which is required for complete combustion.

Flame Check. A gauze, grid, or any other portion of the burner assembly used to avert flash-back.

Flue. The general term for the passages and conduits through which flue gases pass from the combustion chamber to the outer air.

(1) Appliance Flue. The flue passages within an appliance.

(2) Chimney Flue. A conduit for conveying the flue gases delivered into it by a flue pipe, to the outer air.

(3) Flue Pipe (Vent Pipe). The conduit connecting an appliance with the chimney flue.

(4) Dilution Flue. A passage designed to effect the dilution of flue gases with air before discharge from an appliance.

Flue Collar. A projection or recess provided to accommodate the flue pipe.

Flue Gases. Products of combustion and excess air.

Flue Losses. The sensible heat and latent heat above room temperature of the flue gases leaving the appliance.

Flue Outlet (Vent). The opening provided in an appliance for the escape of the flue gases.

Gas Burner Valve. (See Burner Valve.)

Heat Input Rating. The gas burning capacity of an appliance in B.t.u. per hour as specified by the manufacturer.

Heating Surface. All surfaces which transmit heat from flames or flue gases to the medium to be heated.

Heating Value (Total). The number of British thermal units produced by the combustion at constant pressure of one cubic foot of gas (as defined under Cubic Foot of Gas) when the products of combustion are cooled to the initial temperature of the gas and air, when the water vapor formed during combustion is condensed, and when all the necessary corrections have been applied.

Injection Burner. (See Burner.)

Luminous Flame Burner. (See Yellow Flame Burner.)

Main Control Valve. A valve in the gas line before all regulating devices and the branch to the pilot or pilots, except where such pilot or pilots are equipped with independent shut-off valves, for the purpose of completely turning on or shutting off the gas supply to the appliance.

Manifold. The conduit of an appliance which supplies gas to the individual burners.

Mixer. The combination of mixer head, mixer throat, and mixer tube.

(1) Mixer Head. That portion of an injection type burner, usually enlarged, into which primary air flows to mix with the gas stream.

(2) Mixer Throat. The portion of the mixer which has the smallest cross-sectional area and which lies between the mixer head and the mixer tube.

(3) Mixer Tube. The portion of the mixer which lies between the throat and the burner head.

Mixer Face. The air inlet end of the mixer head.

Needle, Adjustable. A tapered projection, coaxial with and movable with respect to an orifice the position of which is fixed, to regulate the flow of gas.

Needle, Fixed. A tapered projection the position of which is fixed, coaxial with an orifice which can be moved with respect to it, to regulate the flow of gas.

Orifice. The opening in an orifice cap, orifice spud or other device whereby the flow of gas is limited and through which the gas is discharged.

Orifice Cap (Hood). A movable fitting having an orifice which permits adjustment of the flow of gas by the changing of its position with respect to a fixed needle or other device.

Orifice Spud. A removable plug or cap containing an orifice. Permits adjustment of the flow of gas either by substitution of a spud with a different sized orifice or by motion of a needle with respect to it.

Pilot. A small flame used to ignite the gas at the main burner or burners.

Port. Any opening in a burner head through which gas or an air-gas mixture is discharged for ignition.

Power Burner. (See Burner.)

Pre-Mixing Burner. (See Burner.)

Pressure Burner. (See Burner.)

Primary Air. The air introduced into a burner which mixes with the gas before it reaches the port or ports.

Primary Air Inlet. The opening or openings through which primary air is admitted into a burner.

Purge. To free a gas conduit of air or mixture of gas and air.

Regulator. A device for controlling and maintaining a uniform pressure on a gas supply.

(1) Spring Type Regulator. A pressure regulator in which the regulating force acting upon the atmospheric side of the diaphragm is derived from a compressed spring.

(2) Dead Weight Type Regulator. A regulator in which the regulating force acting upon the atmospheric side of the diaphragm is derived from a weight or combination of weights.

Riser. Any vertical pipe or nipple which conducts the gas up.

Sag. A low place in a horizontal pipe where liquid may collect.

Secondary Air. The air externally supplied to the flame at the point of combustion.

Specific Gravity. As applied to gas, specific gravity is the ratio of the weight of a given volume of gas to that of the same volume of air, both measured at the same temperature and pressure.

Trap. A section of piping in a horizontal line which is below the general level and from which condensate cannot naturally be removed by grading.

Vent. (See Flue.)

Automatic Stokers. Automatic stokers feed fuel to the furnace exactly as necessitated by the outside temperature and are built for use with either hard or soft coal. Some stokers have a hopper which must be filled manually at intervals, while others take the fuel directly from bins and require no manual filling.
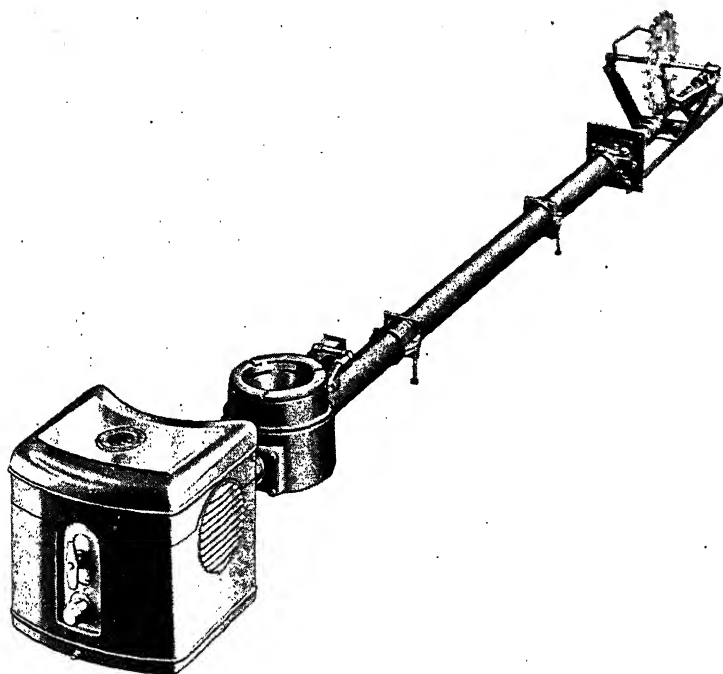
Fig. 164. Iron Fireman Coal Flow, Bin Feed Automatic Stoker for Residence
Courtesy of The Iron Fireman Manufacturing Company, Portland, Oregon

Coal Flow Automatic Stokers. Fig. 164 shows a coal flow type of stoker for use in residences. The principal feature of this stoker is the location of the coal bin on the side of the furnace opposite the fan and drive. The coal is pulled from the bin into the furnace (see Fig. 165). The motor, fan, and gear case are in a direct line with the furnace. The coal flow tube can be installed beneath the floor. In this case, the bottoms of the furnace and motor housings must also be lower than the floor. The length of the coal flow tube varies but the minimum length is generally 5 feet.

The stoker operates as follows: Coal is agitated by the agitator, Fig. 166, and is pulled through the flow tube, Fig. 165, and into the
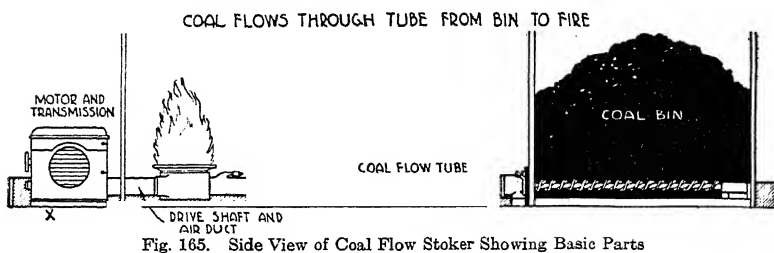
Fig. 165.   Side View of Coal Flow Stoker Showing Basic Parts

retort, Fig. 167. The pulling of coal through the flow tube by a dual pitch feed worm is illustrated in Fig. 168. In the retort, the coal is slowly preheated to the flash point as it moves upward to the top of the fire bed, Fig. 169. The forced draft is supplied by a radial vane fan which is located in the same housing with the motor and transmission. See X in Fig. 165. Volatile gases are liberated in the retort and are completely consumed. The forced draft generates an intense heat. This consumes everything combustible in the coal. The ashes are fused into clinkers which can be lifted out.

Automatic stokers of this type are easily controlled by thermostats as explained in the chapter on "Automatic Controls." They give the full benefit of hourly control, elimination of the fluctuations
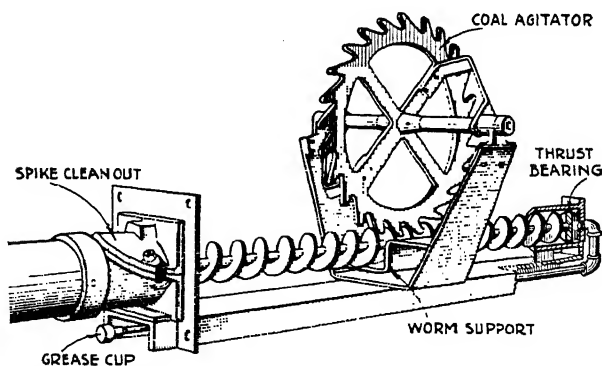
Fig. 166.   Coal Feeding Section of Coal Flow Stoker

of hand firing, and day, night, or week-end temperature regulation.

Other advantages of automatic stokers are fuel economy, smoke elimination, labor saving, and cleanliness. Stokers require the small

size, less expensive coal. Such coal is always easily obtainable. The elimination of smoke means cheaper as well as cleaner heat because
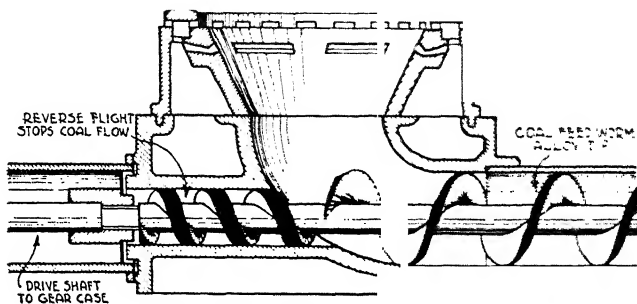
Fig. 167. Cross Section of Retort Showing Forward and Reverse Flights
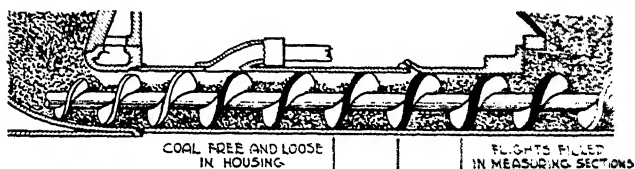
Fig. 168. Cross Section of Flow Tube
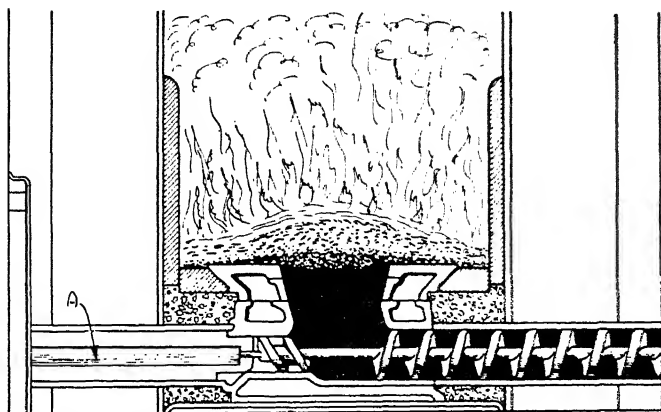
Fig. 169. Cross Section of Retort Showing Fire Bed

there is no wasted fuel where there is no smoke. The labor-saving feature does away with the uncertainties of manual operation. Stoker firing makes no fuel dust and the ashes are in the form of

comparably clean clinkers. This cleanliness makes it possible to use more of the basement area for recreation rooms or children's playrooms, or for space that may be adapted to projects and pursuits of various kinds, such as hobbies, games, collecting, etc.

The selection of a stoker depends on:

(1) Determination of heating load as explained in Chapter II.
(2) Determination of radiation as explained in Chapter VII.
(3) Determination of boiler capacity as explained in Chapter V.

*(4) Selection of stoker to handle boiler load. The explanation for this follows in a form used to determine the size of an Iron Fireman stoker, as shown in Fig. 164. The Heating Survey, pages 293–295, presents data covering stoker selection.

Under General Information the building dimensions (number of rooms, etc.) are used as a check against the radiation. That is, if a very large building is involved with only sufficient radiation for a structure half its size, the survey is rejected until the data is rechecked.

Under Boiler Plant Data all important information is recorded—such as: type of heating system, operating pressure, boiler type, catalogue number, number of sections, stack and breeching dimensions.

When a cast-iron boiler is involved, the †E. D. R. rating should be used as given in the "Net Load, Recommendations for Heating Boilers Catalogue" published by the Heating, Piping and Air Conditioning Contractors' National Association. The boiler rating in E. D. R. is entered in the survey under Recommended E. D. R. Load for Boiler.

Under Domestic Hot Water, if an indirect domestic hot water heater is attached, or is to be attached, to the boiler, the load due to the tank capacity is added to the E. D. R.—see item Number 3, Domestic Hot Water Load, page 294.

The height of the tank above water line and position of tank are also noted.

Under Comparative Operating costs it has been assumed the job is using fuel oil and data pertaining to this fuel has been recorded. To figure comparative operating costs, the following equation is used:

$$\frac{7{,}000 \times 141{,}000 \times 65\% \text{ assumed oil burner efficiency}}{2{,}000 \times \ 14{,}000 \times 60\% \text{ assumed stoker efficiency}}$$

equals 39 tons stoker coal. That is, 39 tons of 14,000 B.t.u. stoker coal will do the job as well as 7,000 gallons of No. 2 oil under similar conditions.

The comparative operating costs are figured from the present fuel cost and cost of available stoker coal and are calculated on the survey sheet. From this, the importance of knowing the stoker coal available, its B.t.u. per pound and cost per ton, is readily apparent. Obviously, these calculations cannot be completed without this information. This is an item frequently overlooked by stoker dealers who should be familiar with available stoker coals.

The steam or hot water radiation is recorded at the top of page 294 in the survey. The total direct radiation increased by one-third for piping and pickup will give the E.D.R. This, plus the load due to heating the domestic hot water, gives the total E.D.R.

*Data Courtesy of The Iron Fireman Manufacturing Co., Portland, Oregon.
†E. D. R.=equivalent direct radiation.

HEATING SURVEY

DOMESTIC HEATING PLANT SURVEY

Dealer __THOMAS COAL CO.__ City __DELROY__ State __WISC.__

Name of Plant __JEFFERSON APTS.__ Address __212 BROAD ST.__ Phone

1. Owner __T. A. MANNING__	2. Buyer	3. Manager __T. A. MANNING.__
Address	Address	Address
4. Engineer	5. Architect	6. Janitor __JOHN HICKS.__
Address	Address	Address

Mail to (1) (2) (3) (4) (5) (6)   Source: Mag. Adv. ☐   Direct Mail ☐   N. P. Adv. ☐   Sales'n ☐   Coal Dealer ☐

GENERAL INFORMATION

Building Measurements: Width __25__ Length __60__ No. Floors __2__ No. Rooms __12__

Type Construction: (Frame, Brick Concrete, etc.) __BRICK__

Water under Concrete? __NO__ Pipes or Drains to interfere with installation? __NO__

Electric Power: D.C. ☐ A.C. ☑ __110__ Volts __60__ Cycles __1__ Phase __Distance Stoker to Suitable Outlet__ __20'__

BOILER PLANT DATA

Steam ☑ Hot Water ☐ Operating Pressure __2__*

Type Boiler __C.I.S.__ Made by __AMER. RAD.__ Catalog No. __3.5.9__ No. C. I. Sections __9__

Size Breeching __15__ Stack Diameter __16x20__ Height __50'__ Give total Boiler H. P. Connected this Stack __GAS WATER HTR.__

Describe in detail condition of Setting and Repair necessary before installing Stoker __BOILER IN__ __GOOD ORDER__

DOMESTIC HOT WATER

Tank Diameter __24"__ Length __5'__ Gal. Capacity __118__ Distance Bottom Water Tank __16"__
(above) (below) Boiler Water line

Size and Type Water Heater __TACO #3A - RUOD GAS__ No. Boiler Sections Tapped __9__

Tank—Ver. ☐ Hor. ☑ No. Bathrooms __4__ Is Present Hot Water Supply Adequate? __YES__

COMPARATIVE OPERATING COSTS

Fuel Used 19__35-36__ Heating Season __7000__ Grade __#2__ Unit Cost __6$__ B.t.u. per Unit __141,000__
(Tons, Gals. M·C·F·)

STOKER COAL	1. Heating only __7000__ @ $ __6$__ Per Unit = $ __420.00__
Trade Name __DENTON__	2. Domestic Hot Water @ $ Per Unit = $
B.t.u. per Pound __14000__	PRESENT TOTAL FUEL COST (1 + 2) = $ __420 00__
Price per Ton $ __6.25__	Less __39__ *Est. Tons Stoker Coal @ $ __6.25__ Per Ton = $ __243.75__
	ESTIMATED ANNUAL FUEL SAVING = $ __176.25__

* If Domestic Hot Water to be Supplied by Stoker, allowance must be made when computing fuel consumption.

Normal reduction in fuel Consumption through stoker firing may in some cases be more than offset by maintaining higher average temperatures in the home over longer periods of time or by heating more rooms than were heated previously.

NOTE: Customary Units Used in Stating Fuel Costs Are: Oil—Gal. (Gallon);   Coal—Tons;
Gas—M.C.F. (Thousand Cubic Feet)

STEAM OR HOT WATER RADIATION

Radiator Type	Height Inches	No. Tubes	No. Sections	Sq. Ft. per Section	Sq. Ft. per Radiator	No. Radiators	Sq. Ft. Radiation
CORTO	32	6	16	5	80	4	320
"	32	6	10	5	50	4	200
"	32	4	6	3½	21	4	84
"	32	5	8	4⅓	34.4	4	137.6
"	32	6	8	5	40	1	40
WALL 7B	21⅞ × 13 5/16				7	15	105

1. TOTAL DIRECT RADIATION.. **886.6**

2. ALLOWANCE FOR PIPING AND PICK-UP .... **886.6** ....(Item 1. Direct Rad.) = **295.5**

 $$\frac{}{3}$$

3. DOMESTIC HOT WATER LOAD.. **118** ....× (1.5 Steam) or (2 Hot Water) = **177.0**
 (Gallons)

EQUIVALENT DIRECT RADIATION = 1 + 2 + 3.. = **1359.1**

RECOMMENDED E.D.R. LOAD FOR **AMERICAN RADIATOR # 3·5·9** ....BOILER = **1658**
 (Lb. per Hour)   (B.t.u. per Lb.)

E.D.R. CAPACITY No. **R·40** ....STOKER = --- **40** .. × . **14000** .. × 0. **60** .Est. Eff. = **1400**
 (240 Steam) or (150 Hot Water)

WARM AIR FURNACE

	RETURN DUCTS				WARM AIR PIPES		
Size Inches	Number	Area Each Pipe	Total Area Square Inches	Diameter Inches	Number	Area Each Pipe	Total Area Square Inches
				8		50.3	
				9		63.6	
				10		78.5	
				12		113.1	
				14		154.0	
TOTAL RETURN DUCT AREA....................							

TOTAL WARM AIR PIPE AREA — SQUARE INCHES =

FURNACE RATED CAPACITY — SQUARE INCHES =

(Pounds per Hour)   (B.t.u. per Pound)

CAPACITY No.......... STOKER = ----------- × ---------- × 0.......... Est. Eff. =
..........(160 Gravity) or (240 Fan)

IMPORTANT: Dimensions should be filled in at each circle.
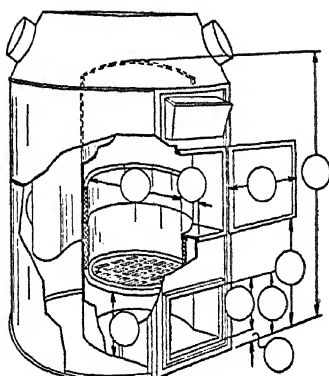
STEEL WARM AIR FURNACE
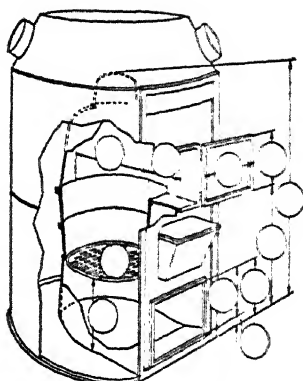
CAST IRON WARM AIR FURNACE
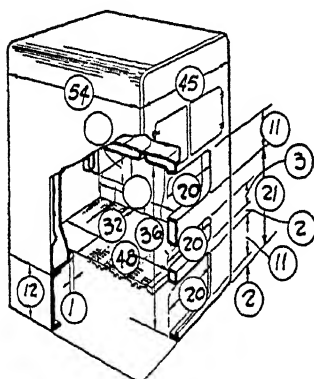
RECTANGULAR BOILER
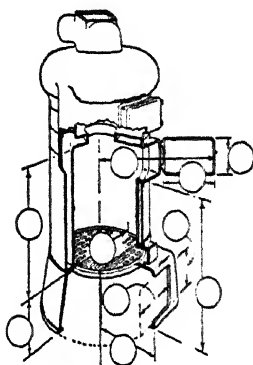
ROUND BOILER

NOTE: Show clearance dimensions around furnace or boiler.
If boiler is now in pit give pit dimensions.

1. MINIMUM HEAD ROOM = $\dfrac{40 \text{ Lb. per Hr.} \times 14,000 \text{ B.t.u. per lb.} \times 12}{**50,000 \qquad X \quad 6.25 \text{ Sq. Ft.}}$ = $31.\frac{1}{2}''$ ....Inches*
(Dead Plate to Boiler Shell)   (Total Grate Area)

HEIGHT DEAD PLATE (Shown in Service Manual)................ = 120 ....Inches
(Allow 2" Base Under Stoker if Possible)

3. (1 + 2) = TOTAL REQUIRED HEIGHT FLOOR TO SHELL.............. = $34.\frac{3}{8}$ ....Inches

4.   PRESENT HEIGHT FLOOR TO SHELL (See Sketch Above).......... = 36 ....Inches

5. (3–4) = MINIMUM PIT DEPTH.. = NONE ....Inches

* The distance from the dead plates to the lowest tube of a Firebox Boiler should be ⅔ Item (1) or greater.
** Where sufficient combustion volume is available behind bridge wall use 75,000 B.t.u. per cubic foot.

RECOMMENDED SETTING: FLOOR ☒ SHALLOW PIT ☐ FULL PIT ☐ FRONT ☒ SIDE ☐
New Boiler Front—Yes ☐ No ☐ Dimensions..   New Fire Doors — Yes ☐ No ☐ Dimensions..

The E.D.R. load is checked with the E.D.R. boiler rating. Where the load is in excess of the rating of a cast-iron boiler, The Iron Fireman Manufacturing Company will not stoker a job.

When the boiler is loaded to only a fraction of its rating, the stoker size selected must handle a rating midway between the load and boiler rating. For example, assume a boiler with 3,000 square feet E.D.R. capacity having a connected E.D.R. load of 1,500 square feet. The stoker capacity is therefore based on 2,250 square feet E.D.R. This is done to have sufficient capacity to heat the boiler quickly and to eliminate the lag in bringing the building up to temperature in a reasonable length of time.

The following equations are used in figuring stoker capacities:

Steam Heating

$$\frac{240 \times \text{E.D.R.}}{\text{B.t.u. per pound stoker coal} \times 60\% \text{ assumed stoker efficiency}} = \text{pounds stoker coal per hour}$$

Hot Water Heating (Gravity Flow)

$$\frac{150 \times \text{E.D.R.}}{\text{B.t.u. per pound stoker coal} \times 60\% \text{ assumed stoker efficiency}} = \text{pounds stoker coal per hour}$$

Warm Air Heating (Gravity Circulation)

$$\frac{160 \times \text{square inches leader pipe area}}{\text{B.t.u. per pound stoker coal} \times 60\% \text{ assumed stoker efficiency}} = \text{pounds stoker coal per hour}$$

Power Job

$$\frac{33,479 \times \text{boiler hp.}}{\text{B.t.u. per pound stoker coal} \times 65\text{--}70\% \text{ assumed stoker efficiency}} = \text{pounds stoker coal per hour}$$

The above calculations are based on standard operating conditions.

For this problem the following equation will determine pounds of stoker coal per hour: $\dfrac{240 \times 1,359.1}{14,000 \times 60\%} = 39$ pounds per hour

From that a 40-pound machine is selected. But the machine will not have sufficient capacity if a lower B.t.u. value coal is used. In territory where coals of various B.t.u. values are commonly used, a 60-pound machine would be installed. However, in Wisconsin where the job is located, a supply of 14,000 B.t.u. fuel is readily available.

In this connection it is also well to take into account the present occupancy of the building, degree-days for the heating season from which the fuel data was recorded, condition of heating plant, etc. If any of these factors are in variance, a 60-pound machine is installed.

For sizing purposes, normally the stoker fired efficiency of boilers is estimated as follows:

Small domestic boilers and furnaces..............................60%
Large *C.I.S. boilers operating somewhat below maximum rating.....65%

*C.I.S.— Cast-iron sectional.

Large C.I.S. boilers operating at, or somewhat near, maximum rating .60%
Steel fire box boilers operating within rating......................65%
Watertube boilers...65%

Data pertaining to Warm Air Furnace installations is recorded at bottom of survey, page 294. The stoker capacity is calculated from square inches of leader pipe area as shown in the preceding equation for that purpose.

It is necessary to have all dimensions of the boiler or furnace to lay out the stoker setting and determine head room from stoker hearth to crown of fire box. These are shown on page 294.

*Pneumatic Spreader Stoker. Fig. 170 shows a pneumatic spreader stoker, which operates on a different principle from the coal
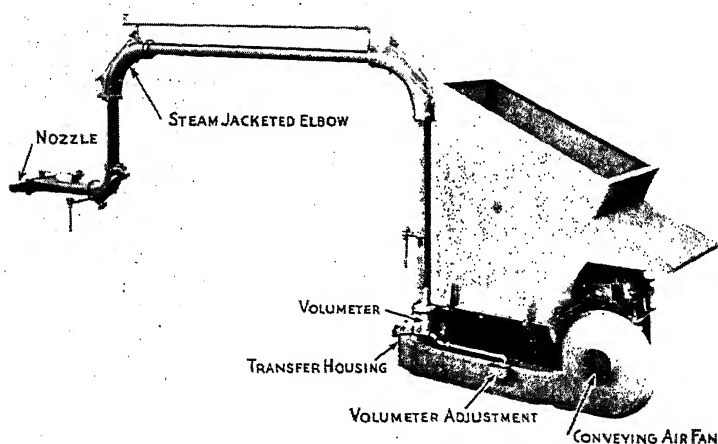
Fig. 170. Pneumatic Spreader Stoker
Courtesy of The Iron Fireman Manufacturing Company, Portland, Oregon

flow type previously described. The coal flow stoker requires a larger size coal than this pneumatic spreader, which uses ordinary commercial screenings or slack coal. The hopper is built of heavy gauge steel. A feed worm measures the coal to provide a constant feed, then carries it from the hopper to the transfer housing. A high-pressure radial vane fan forces air into the housing. There the air stream picks up the coal and carries it through a conveying pipe to the boiler. The slate and other matter heavier than coal fall to the bottom of the transfer housing. They can be removed from there. Steam-jacketed

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*For use with steam boilers.

elbows are provided to prevent damp coal packing in the elbows.
The steam does not enter the coal stream but is piped to and from
the elbows and used only when the coal is unusually damp. A hand
wheel on the outside of the setting elevates the spreader nozzle tip
for front and back distribution.

Fig. 171 illustrates the feeding of coal and air to the fire. The
coal drops from above and the air enters from below. Fig. 172 shows
the type of stoker used for commercial and industrial boilers.
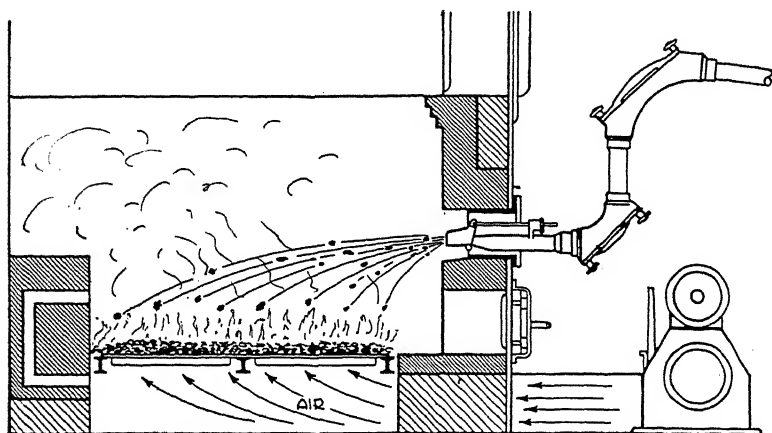
Fig. 171.   Method of Feeding Coal and Air to Fire in Pneumatic Spreader Stoker

*Oil Burners. Oil is the third fuel in general use for automatic
firing. Oil burners, like gas burners, eliminate ashes. Like all auto-
matic equipment, they allow a freedom from the labor of hand firing
and permit an even, controlled temperature.

Oil Fuels. Generally, oil fuels are classified as domestic oils Nos.
1, 2, and 3; and industrial oils Nos. 5 and 6. In some instances, fuels
are referred to as light, medium, and heavy domestic and light,
medium, and heavy industrial oils. Most manufacturers of domestic
burners specify No. 3. Oils of higher numbers contain more water and
solid matter and are more viscous and less volatile. They have higher
flash points. The flash point of an oil is the degree of temperature
at which the application of a torch will cause ignition with a flash.
To determine the flash point, the oil must be slowly heated under

certain definite conditions. Safe limits for domestic fuel oils are 100° to 200°F.* Most industrial oils cannot be used in domestic oil burners. These oils are so viscous they must be preheated and preheating is not a feature of domestic oil burners.

Grades of oil differ in heating value. On a pound basis, light oils have a greater heating value than heavy oils. However, fuel oils are sold by the gallon, and on this basis the heavier oils contain a greater number of heat units per gallon than do the light oils. Table 73,
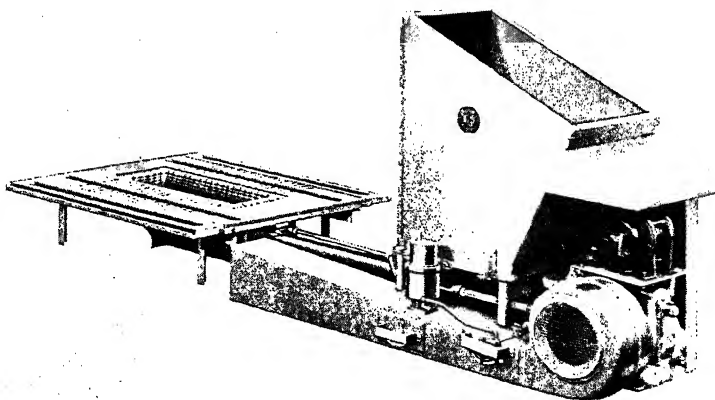
Fig. 172.   Iron Fireman Commercial-Industrial Stoker
Courtesy of The Iron Fireman Manufacturing Company, Portland, Oregon

page 301, shows approximate average heat values of various oil fuels for domestic use.

Types of Oil Burners. Many varieties of oil burners similar to that in Fig. 173 are being manufactured for use in home heaters. All have the primary function of either vaporizing or atomizing fuel oil and mixing it with air for proper combustion. There are, then, two general classes of burners—in addition to the individual blue-flame burner which combines vaporization with atomization.

At the present time, there are two principal types of vaporizing oil burners. The first of these mixes air and oil vapor before combus-

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*Commercial standard specifications for fuel oils can be obtained from the Division of Trade Standards, U. S. Bureau of Standards, Washington, D. C.

tion by means of a mixing chamber or retort and a system of air distribution. Fig. 174 shows this type of burner. In this burner, a mechanism consisting of a blower, motor, and oil-control valves, most of which are located under the hood h, deliver the oil and air
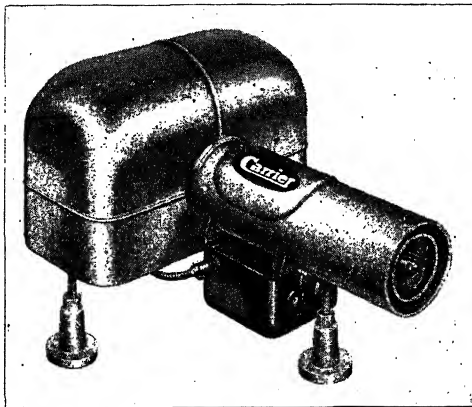
Fig. 173. Typical Oil Burner for Domestic Use
Courtesy of Carrier Engineering Corporation, Newark, N. J.

to the combustion chamber. The air goes to the mixing chamber through the delivery tube g and the oil goes to the chamber through the oil line f. The gas-pilot tip e provides the ignition on the gas-ignition type burner illustrated. Some burners are manufactured with electric ignition. The oil flowing to the bottom of the combustion chamber d is vaporized and premixed with air supplied positively by the blower, principally through the air distributor b. A blower or
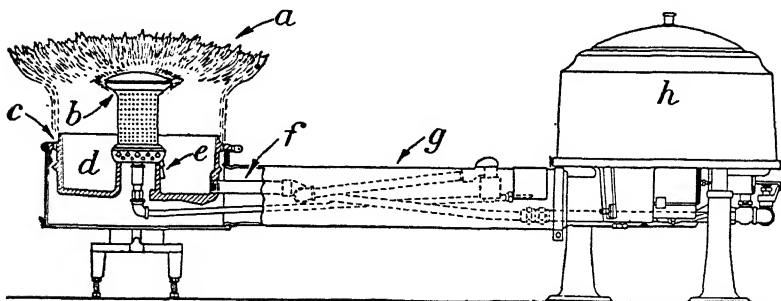
Fig. 174. Pot-type Vaporizing Burner: a, flame; b, air distributor; c, ports; d, combustion chamber; e, gas-pilot tip; f, oil line; g, delivery tube; h, hood
Courtesy of U. S. Department of Agriculture

pump regulates the supply of air to bring about the proper mixture of air and fuel. This type of burner is best known to the general public as the gun type. (The term "pressure-atomizing" burner would be more exact and it is probable that in the future this term will be used more commonly.) Use of the blower does not necessarily cause excessive amounts of heated gases to be blown through the boiler and up the stack.

The operation of the gun-type burner is as follows: Oil goes to the burner from the oil-storage tank through the line, passes through the filter and thence to the oil pump. The pump forces the oil through the pressure-regulating valve and from there the oil is delivered at a constant pressure. Generally there is a so-called "high-low" control of the flame which results in more nearly continuous operation.

The atomizing type may be illustrated by a typical application made by the General Electric Company. In Fig. 175 is shown the equipment for preparing and burning the oil. Air collides with oil inside a mixing chamber. The result is an extremely fine emulsification of air and oil—comparable to beaten egg white. Thus the oil is broken up and intimately mixed with the air necessary to combustion.

Table 73. *Heat Content of Oil Fuels

Oil	Heat Units		Oil	Heat Units	
	Per Lb. B.t.u.	Per Gal. B.t.u.		Per Lb. B.t.u.	Per Gal. B.t.u.
Kerosene	20,000	136,000	No. 2 fuel oil	19,700	140,000
No. 1 fuel oil	19,850	137,000	No. 3 fuel oil	19,500	141,000

*Courtesy U. S. Department of Agriculture.

Next, the emulsified mixture of oil and air is discharged through an orifice into a large combustion chamber which is under less than atmospheric pressure. The air tends to expand and each small bubble of the emulsion bursts into millions of particles. Fine oil ignites easily. It is done in this case with a 12,000 volt spark (1,000°–1,500°F.). Oil normally ignites at 600°F. As each bubble passes over the spark, a halo of gas forms about it. This gas is raised to kindling temperature and takes fire. The flames are conical in shape and burn on the outside surface. The heat of the flame changes into gas the atomized oil inside the cone. However, the amount of air coming in

contact with it is limited and there is no combustion until the atom-
ized oil has been converted. At the right moment a second supply of
air is introduced at the bottom. Combustion takes place without
forming soot. This secondary air supply from the bottom turns the
flame back upon itself and tends to double effectiveness.
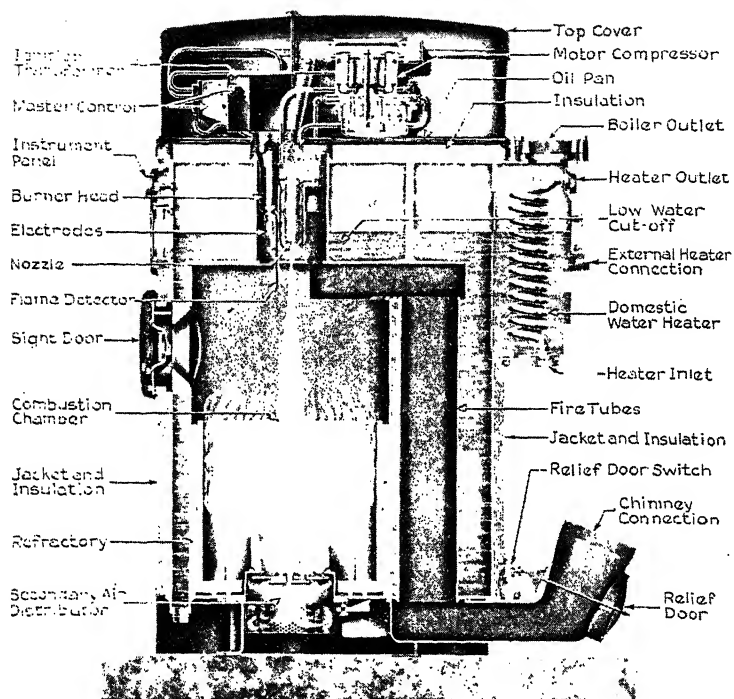
Fig. 175. Oil Furnace Sectioned to Show the Equipment for Preparing and Burning Oil. This
type is used for domestic steam, vapor, hot water, and air-conditioning systems
Courtesy of General Electric Company, Bloomfield, N. J.

General Ignition Methods. The electric arc method of ignition,
shown in Fig. 175, has been largely used. The two wires on the left
of the bottom portion of the nozzle are the electrodes which provide
the intermittent spark to ignite the mixture. Where gas is used,
a pilot maintains a steady heat. It must be placed so it can bring the
mixture of oil and air to a combustion temperature most effectively.
Other principles are used in the ignition of the mixtures but electric
ignition is fast becoming the most popular.

Selection of Oil Burners. The heating apparatus is the first item to consider in selecting an oil burner. A boiler or furnace must be of ample size to obtain good results with any kind of burner. Heating systems should be designed carefully—following recommended codes for warm air furnaces, and carefully selecting boilers and radiators for steam or hot water systems.

Oil burners selected must be of ample capacity to provide for maximum demands. The practice of manufacturers in supplying heating capacities varies widely. Some make a series of burners of similar construction but of different size and oil capacity. Some make two or three types of burners, each of which can be adjusted to fit varying demands. Other manufacturers make only one type of burner and depend on adjustments to adapt the burner to demands.

During severe winter weather, at least $\frac{1}{5}$ gallon of oil per hour is required for each 100 square feet of hot-water radiation. Approximately $\frac{1}{3}$ gallon is required for each 100 square feet of steam radiation. These figures do not allow for the use of a domestic hot water supply. Burners should be selected to satisfy peak demands.

The United States Department of Agriculture has found it is impossible to give a general answer as to what type of burner will operate best with given boilers. However, some combinations of boilers and burners are better than other combinations. Vertical-rotary burners do best in round boilers. Pot- or gun-type burners can generally be used with either round, square, or elongated fire pots. The vertical-rotary or pot-type burners are more efficient than the gun-type burners in boilers which are deficient in flue travel.

Converting Existing Heating Plants into Oil Burning Plants. Thought must be given to the inner surface of the fire box of an existing boiler or furnace when considering its conversion. In most cases these surfaces are either iron or steel. Both have a high heat transfer rate. This may arrest combustion and cause a soot deposit in the heating passages which will seriously reduce heating efficiency. Complete combustion requires high heat levels and the rapid transfer through the fire box surfaces lowers the heat level. To overcome this, some fire boxes can be made more suitable for oil burning if they are lined with a form of fire brick which has a low heat transfer.

The proper shape burner for a warm air furnace can be selected without trouble. Most furnaces are round, so round burners may be

used, or the specifications given previously may be followed. The joints of cast-iron fire pots sometimes cause difficulty. An oil burner is controlled by a thermostat which turns the burner off and on. Thus the burner is either burning at top capacity or not burning at all. This alternation of heating and cooling loosens seams. This, in turn, allows the products of combustion to enter the warm air supply and reach the living quarters of the house. This difficulty can be avoided by partially dismantling the furnace, locating the leaks, and sealing them in the proper manner. Steel furnaces are not subject to such troubles.

An insufficient combustion space, heat-absorbing surface, or inadequate gas travel in an existing steam or hot-water boiler may cause difficulty during conversion. A remedy for such a condition is the addition of another section where structural details of the building, such as head room, permit.

Some simple and inexpensive schemes can be used with otherwise inefficient boilers to effect material reductions in fuel consumption. One such expedient consisting of a baffle is shown in Fig. 176. In this four-section round boiler there is very little flue travel and therefore very small opportunity for absorption of heat by the water. This type of boiler with an oil burner wastes heat up the chimney. The simplest scheme for reducing such waste is the installation of a baffle. The form illustrated consists of a sheet of metal supported on split fire brick or similar material in such a manner as to cause the hot gases to take the path indicated rather than to short-circuit directly from one opening to another. The baffle causes the gases to **scrub** the heat-absorbing surface more effectively.

The view shows the clearance that must be allowed between the circumference of the baffle and the boiler wall to permit passage of the gases. This opening must not be too small or a back pressure will be built up in the combustion chamber.

Such baffling should be done by the burner agent. He should use a draft gauge to determine the draft in the combustion chamber, to avoid excessive stoppage of the flow of the products of combustion. Although oil-burner men are familiar with this or other forms of baffles, not many of them are aware of the appreciable reductions in stack temperature and increases in economy that are possible through their use. Tests by the Department of Agriculture show that the use

of such a simple baffle in a four-section 20-inch round boiler, in doing the same heating job, resulted in a decrease of approximately 125°F. in stack temperature and a saving of fuel of approximately 8%. Different types of burners lend themselves to different degrees of baffling. The gun-type, retort- or pot-type, and the yellow-flame vertical-rotary burner can withstand the effects of considerable
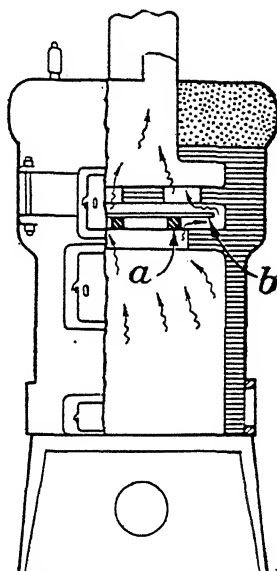

Fig. 176. Four-Section Round
Boiler Showing Use of Baffles:
a, split brick; b, baffle
Courtesy of
U. S. Dept. of Agriculture

baffling, but with the blue-flame rotary burner excessive back pressure often results in ignition difficulties.

Several oil-burner manufacturers have for some time been making equipment to be used in connection with existing boilers to improve their efficiency by providing additional heating surfaces.

Burner Adjustment. After a burner has been installed in a boiler, it must be properly adjusted for efficient service. The adjustment consists of regulating the quantities of oil and air admitted. The rate of oil consumption depends on the heating load; that is, on the size, arrangement, and construction of the house, the room temperature to be maintained, the outdoor temperatures, the exposure

to winds, etc. The adjustment of the air admission depends upon the quantity of fuel to be burned, for a definite quantity of fuel burned requires a definite quantity of air.

Soot and smoke generally result if insufficient air is supplied. It is not easy to supply precisely the amount of air theoretically needed for perfect combustion. Even if this were accomplished the intermingling of the oil and air probably would not be sufficiently complete to give perfect combustion, so an excess of air is necessary to ensure that each subdivided bit of oil is provided with the amount of air needed. In practice possibly 25 to 50 per cent more air is supplied than is theoretically required. An excess of air is also advantageous with automatic operation when the burner is started and stopped frequently, because it lessens the smoky condition which sometimes occurs when the burner comes on.

In general it can be said that introduction to the furnace of more air than is necessary for combustion reduces flame temperature and results in otherwise useful heat being carried up the chimney. It is not easy, however, for the layman to determine how much excess air is being supplied to his burner; this can be determined only by an apparatus known as a flue-gas analyzer.

Today a large percentage of oil-burner installations include a so-called automatic draft control of some description, which helps to maintain the adjustment of the burner. This generally consists of a swinging check damper which automatically opens or closes to hold a predetermined draft intensity at the furnace. The intensity of this draft usually can be altered by moving a balance weight. Because such a device almost always helps burner operation and is low in cost, it might well be part of every burner installation. A slightly different automatic draft control acts as an ordinary shut-off damper instead of as a check damper, with probably the same principal effect.

It is of interest to note the large quantities of air required to support combustion in house heating. The oil fuels now employed are very uniform in composition and contain roughly 84% carbon and 13% hydrogen, oxygen, nitrogen, and sulphur together composing the other 3%. One gallon of such oil will require about 2,000 cubic feet of air for combustion. Thus, if a burner is using 2 gallons of oil per hour, the air required each hour will be approximately 4,000 cubic feet, which is the equivalent of the entire content of a basement

20 feet square and 10 feet high. Usually no special openings into the basement are necessary, as enough air will come in by infiltration. Only in the case of a small basement of tight construction should any special openings be provided.

Table 74 gives typical ratings for oil burners.

Table 74. S. T. Johnson Co. Rotary Oil Burners Types 28, 30-AV, 30-H

Burner Size No.	Capacity Steam *Radiation, Sq. Ft.		Boiler Hp. Rating	Gal. Oil per Hour		Size Motor Hp.	Dimensions in Inches					
	Min.	Max.		Min.	Max.		A	B	C	D	E	F
2½	825	2775	20	2	7	⅓	14	13⅞	9½	6¹⁵⁄₁₆	17	15
3½	1400	6950	50	3	15	½	16³⁄₁₆	15⅞	10¼	8¼	17½	15½
4½	2775	13900	100	6	30	¾	17	19½	13	9⁵⁄₁₆	21	19¼
5½	5550	27800	200	12	60	2	21½	21½	13	11	24	20
6½	8325	41700	300	24	90	3	21½	21½	13	11	25	20½

*Boiler output. Above burners available without built-in pumps in same sizes and capacities.
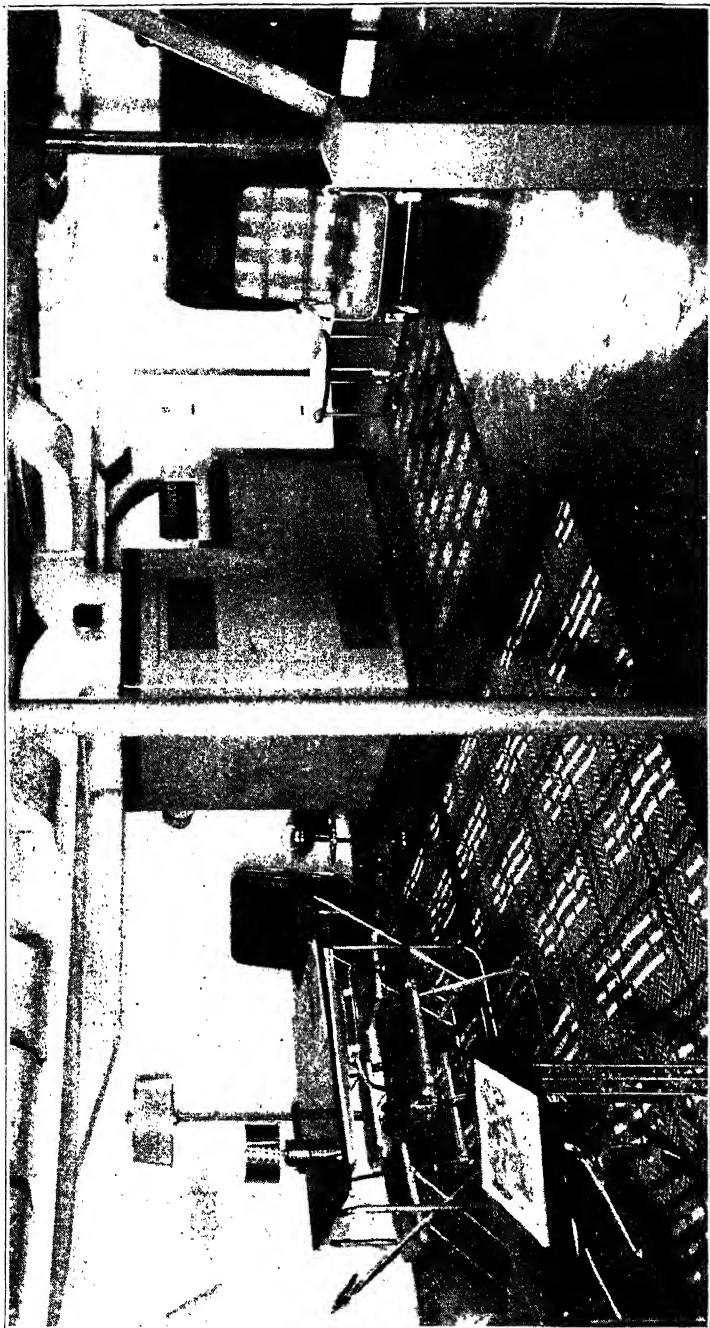
A CAREFULLY PLANNED DUCT LAYOUT

Turns and offsets are held to a minimum to avoid unsightliness.

Courtesy of Gilbert & Barker Co., Springfield, Mass.

CHAPTER XVI

AIR=CONDITIONING APPLIANCES

Air Washers. Fig. 177 shows an air washer typical of those used in central system air-conditioning installations. An air washer is essentially an enclosure where air is brought into close contact with water and where the objective is either simply to wash the air or to wash the air and regulate its temperature and moisture content. The air contacts the water by means of sprays of the latter or by passing over and around wet surfaces or by a combination of both methods.
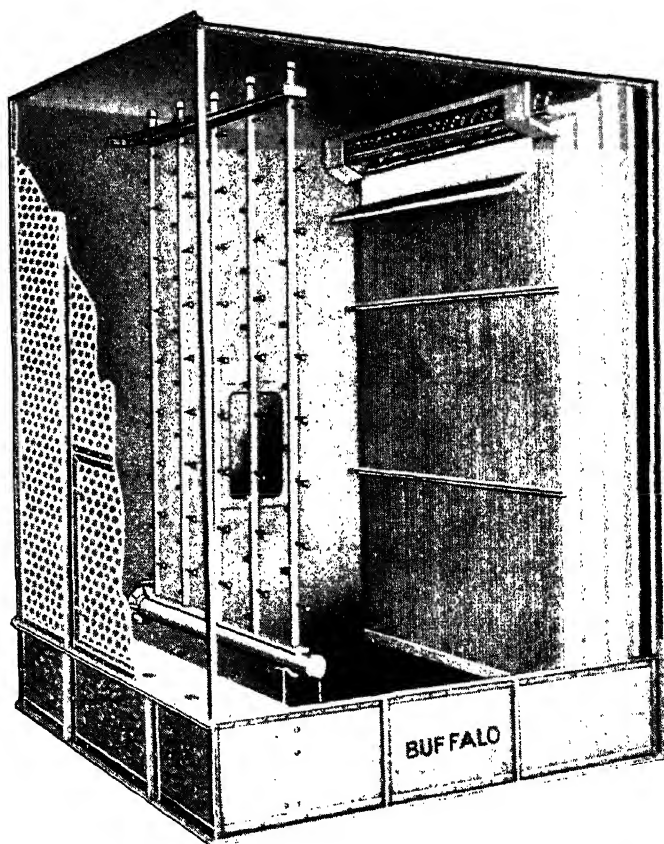
Fig. 177. Air Washer with Side and Distributing Plate Removed to Show Interior
Courtesy of Buffalo Forge Company, Buffalo, N. Y.

In the spray method a number of nozzles, Fig. 178, which spray the water, are placed in a bank, Fig. 177, across the path of the air and the water is forced through them by a pump. The most common
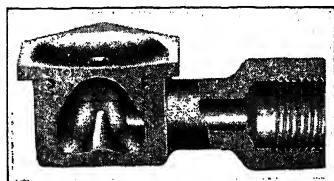
Fig. 178. Section of Spray Nozzle
Courtesy of Buffalo Forge Company,
Buffalo, N. Y.

washer consists of spray nozzles, spray chamber, and eliminators. When the water is forced through the spray nozzles, it forms a mist. Where necessary, two or more banks of spray nozzles may be used, as in Fig. 179. When the fan draws the air through the mist, some of the dirt, gases, etc., are removed. The eliminator plates, Fig. 180, do most of the actual cleaning of the air. The eliminators follow the
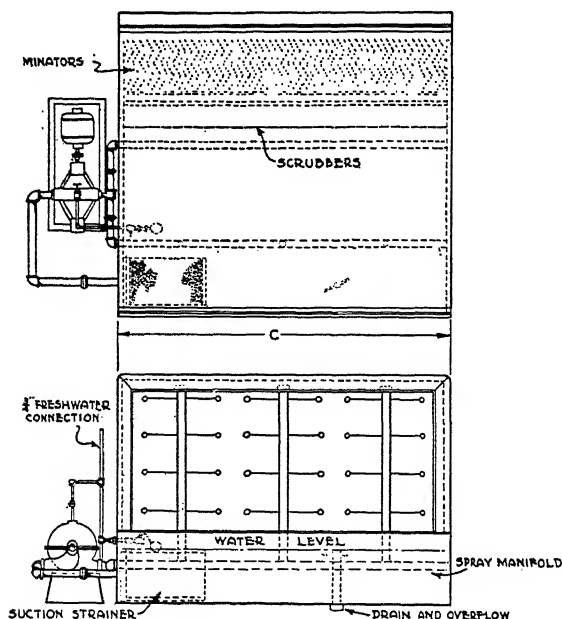
Fig. 179. Typical Two-Bank Air Washer

spray in sequence and are so designed that they suddenly change the
direction of the air and cause the dirt to be thrown out of the air by
inertia. In addition, the air *scrubs* the surfaces of the eliminators,
which being wet, tend to remove dirt from the air. The eliminators
are supplied with a rain-like supply of water which keeps them
washed clean at all times. The last few corrugations of the elimi-
nators effectively remove all free or unabsorbed moisture, so it cannot
be carried to the conditioned rooms or enclosures.
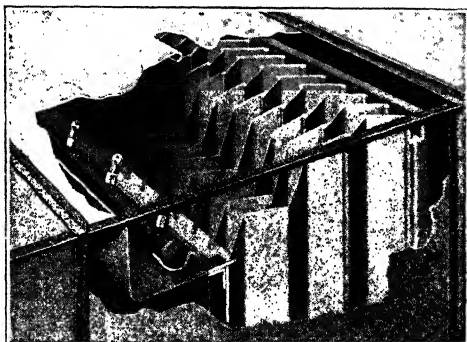
Fig. 180.   Part of Casing Removed Showing Eliminators
in Position
Courtesy of Buffalo Forge Company, Buffalo, N. Y.

All washers have a tank constructed in their lower parts, as shown
in Fig. 179, to catch the falling water. The circulating pump takes
its supply from this tank and creates recirculation. A screen, Fig.
179, is installed on the suction line to prevent circulation of the dirt
that accumulates in the tank. The dirt in the tank should be re-
moved frequently.

If freezing air is being drawn into the washer, a tempering coil,
as shown in Fig. 41 in Chapter VI on "Ventilating Systems," is neces-
sary to keep the water from freezing.

To obtain best washer performance, air should enter the spray
chamber and should be distributed evenly over the washer inlet.
To make this possible, perforated plates or eliminator plates can be
installed at the inlet. This also prevents water from the sprays
escaping from the spray chamber.

Some washers have a second set of eliminators or scrubbers.
This type is used where fine dust, etc., must be eliminated from the

air—as in industrial plants. In practice, there are two additional types of air washers. One is called the *A* type and is used primarily for humidifying in the winter months. Such a washer generally contains a single bank of sprays. The other is called the *B* type and is used primarily as a dehumidifier in the summer months. This type generally has two banks of sprays.

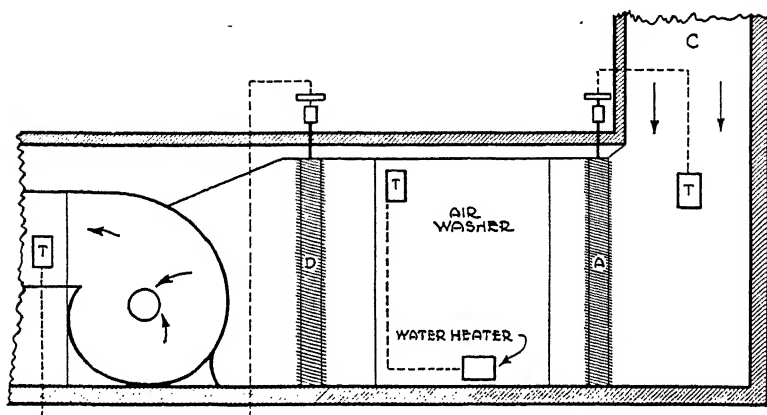*Heating and Humidifying.* During the winter months the air washer is used only for cleaning and humidifying the air. Fig. 181

Fig. 181. Arrangement of Apparatus

shows a typical system in which all air is exhausted and none recirculated. The air enters at *C* from an outside source. It goes through preheater coil *A* whose function it is to warm the incoming air to a point where it will not freeze the water in the washer. The thermostat in the fresh air supply duct starts or stops the steam supply to preheater coil *A* as required by the temperature of the incoming air. Next, the air (temperature approximately 30°F. when outside temperature is 0°F.) is drawn into the air washer. Here the washing is done as previously explained. To humidify the air, the water in the air washer is heated. Fig. 181 shows the heater, which may be located within or near the washer, and the thermostat which controls the operation of the heater to keep the water at the required temperature. Thus the air is heated before entering the washer and the spray water is heated also. The conditions are then right for adding humidity to the air. When ample heat is provided in this manner, the air leaves the washer clean and almost saturated.

From the air washer the air is drawn through reheater coil D where it is given the final heat to provide air at the proper temperature. The thermostat in the duct beyond the fan controls the supply of steam to reheater coil D.

Various systems similar to that illustrated in Fig. 181, with the exception of by-passing and recirculation, are shown and explained in Chapter VI on "Ventilating Systems."

Automatic control systems for typical systems using air washers are shown in Chapter XII on "Automatic Controls."

Cooling and Dehumidifying. During the summer months when washing, cooling, and dehumidification are required, the air washer, such as shown in Fig. 181, will give the desired results. The coils A and D will be inoperative. The washing operation is the same as for winter operation.

The water in the air washer is sprayed as in winter, except that usually two banks of spray nozzles are used. The water is used for the heat transfer or exchange for the lowering of the temperature and the removal of moisture vapor of the air. This process requires the use of water in the sprays at temperatures low enough to cool and dehumidify. The water can be taken from city supplies when its temperature is below 70°F., from wells, or other low temperature water sources. The use of a refrigerant to cool the water is resorted to only when the water is not naturally cold enough.

Before moisture can be removed from the air, its temperature must be lowered below the dew point temperature. (The dew point temperature is that at which dew is formed or vapor is precipitated.) If entering air contains more moisture than is desired, the function of the air washer is to cool this air sufficiently to "squeeze out" the excess moisture. Saturated air is 100% humidified and can carry no more vapor. The saturation point of air varies with temperature and if air is cooled several degrees it will lose some of its moisture. Thus cooling the air lowers its moisture content or dehumidifies.

Sometimes, in order to reach the required degree of dehumidification, the air must be cooled below the temperature desired in the areas being conditioned. This is offset by operating a heating coil such as D in Fig. 181.

General Principles. The prevailing idea that the application of proper humidification of interior spaces results in a saving of heat is

erroneous. The amount of heat required for transformation of the moisture to a vapor constituent of the air is equal to the amount of heat saved by the lower temperature maintained in the occupied spaces. To change one pound of moisture to a vapor, at atmospheric pressure, requires 1,000 to 1,100 B.t.u.'s per pound. This heat transfer will vary according to the temperature of the air and the moisture or water in service.

The amount of water for air washer requirements in humidifying, dehumidifying, cleaning, etc., is based on the number of gallons per 1,000 cubic feet of air per minute according to the use of the sections of the washer. Roughly, the water requirements average 4 to 8 gallons of water per minute per 1,000 cubic feet of air per minute. A water pressure of about 30 pounds per square inch is ample.

The temperature and humidity of the air are closely related and both contribute to the comfortable occupation of structures. Variations of the air temperature and vapor mixture cause a change of the volume of the air. If the mixture is reduced in temperature without any loss or removal of moisture, the air will eventually become saturated. Further reduction of the temperature of the air and vapor will result in a dew formation and a reduction of the volume of air and vapor mixture.

If the air and vapor mixture is increased in temperature, the volume of the air and the capacity for vapor are also increased. Thus the ratio of the moisture vapor actually in the air to the moisture vapor that air is able to carry is lowered. That is, the relative humidity is lowered.

The relative and the absolute humidities will vary in the interior and exterior spaces. The relative humidity of interior spaces is high in the morning, lower during the midday period, and high again during the evening or night period. The average relative humidity for the outdoors is 70 to 75%. The average indoor relative humidity should be maintained between 40 and 50%.

High relative humidities in winter should not be attempted unless the structure in question has been thoroughly insulated. This means all surfaces which actually come in contact with outdoor temperatures. Windows should be double glazed and doors protected as much as possible. Without thorough insulation, condensation would occur and ruin woodwork, plaster, floors, interior decoration, etc.

Table 75. *Mixtures of Air and Saturated Water Vapor

Temp. °F.	Weight of Saturated Vapor		Volume in Cu. Ft.		Heat Content in B.t.u. of 1 Lb. of Dry Air Above 0°F.	Latent Heat of Vapor B.t.u.	†Heat Content in B.t.u. of 1 Lb. of Dry Air with Vapor to Saturate It
	Per Cu. Ft. Grains	Per Lb. of Dry Air Grains	Of 1 Lb. of Dry Air	Of 1 Lb. of Dry Air + Vapor to Saturate It			
0	0.472	5.47	11.58	11.59	0.0	0.964	0.964
2	.522	6.08	11.63	11.65	0.482	1.071	1.553
4	.576	6.74	11.68	11.70	0.964	1.186	2.150
6	.636	7.47	11.73	11.75	1.446	1.313	2.759
8	.701	8.28	11.78	11.80	1.928	1.455	3.383
10	0.772	9.16	11.83	11.86	2.411	1.608	4.019
12	.850	10.13	11.88	11.91	2.893	1.776	4.669
14	.935	11.19	11.94	11.97	3.375	1.961	5.336
16	1.028	12.35	11.99	12.02	3.858	2.162	6.020
18	1.128	13.62	12.04	12.08	4.340	2.383	6.723
20	1.237	15.01	12.09	12.13	4.823	2.623	7.446
22	1.356	16.52	12.14	12.19	5.305	2.885	8.190
24	1.485	18.17	12.19	12.24	5.787	3.170	8.957
26	1.625	19.98	12.24	12.30	6.270	3.482	9.752
28	1.776	21.94	12.29	12.35	6.752	3.821	10.573
30	1.943	24.11	12.34	12.41	7.234	4.195	11.429
32	2.124	26.47	12.39	12.47	7.716	4.058	11.783
33	2.206	27.57	12.41	12.49	7.96	4.22	12.18
34	2.292	28.70	12.44	12.52	8.20	4.40	12.60
35	2.380	29.88	12.47	12.55	8.44	4.57	13.02
36	2.471	31.09	12.49	12.58	8.68	4.76	13.44
37	2.566	32.35	12.52	12.61	8.93	4.95	13.87
38	2.663	33.66	12.54	12.64	9.17	5.14	14.31
39	2.764	35.01	12.57	12.67	9.41	5.35	14.76
40	2.868	36.41	12.59	12.70	9.65	5.56	15.21
41	2.976	37.87	12.62	12.73	9.89	5.78	15.67
42	3.087	39.38	12.64	12.76	10.14	6.01	16.14
43	3.201	40.93	12.67	12.79	10.38	6.24	16.62
44	3.319	42.55	12.69	12.82	10.62	6.48	17.10
45	3.442	44.21	12.72	12.85	10.86	6.73	17.59
46	3.568	45.94	12.74	12.88	11.10	6.99	18.09
47	3.698	47.73	12.77	12.91	11.34	7.26	18.60
48	3.832	49.58	12.79	12.94	11.58	7.54	19.12
49	3.970	51.49	12.82	12.97	11.83	7.83	19.65
50	4.113	53.47	12.84	13.00	12.07	8.12	20.19
51	4.260	55.52	12.87	13.03	12.31	8.43	20.74
52	4.411	57.64	12.89	13.07	12.55	8.75	21.30
53	4.568	59.83	12.92	13.10	12.79	9.08	21.87
54	4.729	62.09	12.95	13.13	13.03	9.41	22.45
55	4.895	64.43	12.97	13.16	13.28	9.76	23.04
56	5.066	66.85	13.00	13.20	13.52	10.13	23.64
57	5.242	69.35	13.02	13.23	13.76	10.50	24.25
58	5.424	71.93	13.05	13.26	14.00	10.89	24.88
59	5.611	74.60	13.07	13.30	14.24	11.28	25.52
60	5.804	77.3	13.10	13.33	14.48	11.69	26.18
61	6.003	80.2	13.12	13.36	14.72	12.12	26.84
62	6.208	83.2	13.15	13.40	14.97	12.56	27.52
63	6.418	86.2	13.17	13.43	15.21	13.01	28.22
64	6.633	89.3	13.20	13.47	15.45	13.48	28.93
65	6.885	92.6	13.22	13.50	15.69	13.96	29.65
66	7.084	95.9	13.25	13.54	15.93	14.46	30.39
67	7.320	99.4	13.27	13.58	16.18	14.97	31.15
68	7.563	103.0	13.30	13.61	16.42	15.50	31.92
69	7.813	106.6	13.32	13.65	16.66	16.05	32.71
70	8.069	110.5	13.35	13.69	16.90	16.61	33.51
71	8.332	114.4	13.38	13.73	17.14	17.19	34.33
72	8.603	118.4	13.40	13.76	17.38	17.79	35.17
73	8.882	122.6	13.43	13.80	17.63	18.41	36.03
74	9.168	126.9	13.45	13.84	17.87	19.05	36.91

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†Values in this column do not include the heat of the liquid. Below 32°F. the heat of sublimation of ice is included.

*Table 75—Continued

Temp. °F.	Weight of Saturated Vapor		Volume in Cu. Ft.		Heat Content in B t.u. of 1 Lb. of Dry Air Above 0°F.	Latent Heat of Vapor B.t.u.	†Heat Content in B.t.u. of 1 Lb. of Dry Air with Vapor to Saturate It
	Per Cu. Ft. Grains	Per Lb. of Dry Air Grains	Of 1 Lb. of Dry Air	Of 1 Lb. of Dry Air + Vapor to Saturate It			
75	9.46	131.4	13.48	13.88	18.11	19.71	37.81
76	9.76	135.9	13.50	13.92	18.35	20.38	38.73
77	10.07	140.7	13.53	13.96	18.59	21.08	39.67
78	10.39	145.6	13.55	14.00	18.84	21.80	40.64
79	10.72	150.6	13.58	14.05	19.08	22.55	41.63
80	11.06	155.8	13.60	14.09	19.32	23.31	42.64
81	11.40	161.2	13.63	14.13	19.56	24.11	43.67
82	11.76	166.7	13.65	14.17	19.80	24.92	44.72
83	12.12	172.4	13.68	14.22	20.04	25.76	45.80
84	12.50	178.3	13.70	14.26	20.29	26.62	46.91
85	12.89	184.4	13.73	14.31	20.53	27.51	48.04
86	13.28	190.6	13.75	14.35	20.77	28.43	49.20
87	13.68	197.0	13.78	14.40	21.01	29.38	50.39
88	14.10	203.7	13.80	14.45	21.25	30.35	51.61
89	14.53	210.6	13.83	14.50	21.50	31.36	52.86
90	14.	217.6	13.86	14.55	21.74	32.39	54.13
91	15.41	224.9	13.88	14.60	21.98	33.46	55.44
92	15.87	232.4	13.91	14.65	22.22	34.59	56.78
93	16.34	240.1	13.93	14.70	22.46	35.69	58.15
94	16.82	247.1	13.96	14.75	22.71	36.86	59.56
95	17.32	256.3	13.98	14.80	22.95	38.06	61.01
96	17.82	264.8	14.01	14.86	23.19	39.30	62.48
97	18.35	273.6	14.03	14.91	23.43	40.57	64.00
98	18.88	282.5	14.06	14.97	23.67	41.88	65.55
99	19.42	291.8	14.08	15.02	23.91	43.24	67.15
100	19.98	301.3	14.11	15.08	24.16	44.63	68.79
101	20.56	311.2	14.14	15.14	24.40	46.07	70.47
102	21.15	321.4	14.16	15.20	24.64	47.54	72.18
103	21.75	331.9	14.19	15.26	24.88	49.07	73.95
104	22.36	342.7	14.21	15.33	25.13	50.64	75.77
105	22.99	354	14.24	15.39	25.37	52.26	77.63
106	23.64	365	14.26	15.46	25.61	53.92	79.53
107	24.30	377	14.29	15.52	25.85	55.64	81.49
108	24.98	389	14.31	15.59	26.09	57.41	83.50
109	25.67	402	14.34	15.66	26.33	59.23	85.57
110	26.38	415	14.36	15.73	26.58	61.11	87.69
111	27.11	428	14.39	15.80	26.82	63.04	89.86
112	27.85	442	14.41	15.87	27.06	65.04	92.10
113	28.61	456	14.44	15.95	27.30	67.10	94.40
114	29.39	471	14.46	16.02	27.55	69.22	96.77
115	30.18	486	14.49	16.10	27.79	71.40	99.10
116	31.00	502	14.52	16.18	28.03	73.65	101.68
117	31.83	518	14.54	16.26	28.27	75.97	104.24
118	32.68	534	14.57	16.35	28.51	78.36	106.87
119	33.55	551	14.59	16.43	28.76	80.80	109.56
120	34.44	569	14.62	16.52	29.00	83.37	112.37
125	39.19	667	14.75	16.99	30.21	97.33	127.54
130	44.49	780	14.88	17.53	31.42	113.64	145.06
135	50.38	913	15.00	18.13	32.63	132.71	165.34
140	56.91	1072	15.13	18.84	33.85	155.37	189.22
145	64.1	1260	15.26	19.64	35.06	182.05	217.1
150	72.1	1485	15.39	20.60	36.27	214.03	250.3
155	80.9	1758	15.52	21.73	37.48	252.61	290.1
160	90.6	2091	15.64	23.09	38.69	299.55	338.2
165	101.1	2504	15.77	24.75	39.91	357.75	397.7
170	112.8	....	15.90	26.84	41.12	431.2	472.3
175	125.5	....	16.03	29.51	42.33	526.0	568.3
180	139.4	....	16.16	33.04	43.55	651.9	695.5
185	154.4	....	16.28	37.89	44.76	826.1	870.9
190	170.9	....	16.41	45.00	45.97	1082.3	1128.3
200	208.0	....	16.67	77.24	48.40	2247.5	2296

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†Values do not include heat of liquid. Below 32°F. heat of sublimation of ice is included.

Formulas. The application of humidification, dehumidification, etc., is governed by calculations involving the use of the Psychrometric Chart found in the back of the book and also Table 75 for "Mixtures of Air and Saturated Water Vapor." Many formulas are also necessary, which are given in the following.

Humidification. Formulas for calculating grains of vapor necessary to be added to air by humidifier for humidification during the heating season are as follows:

$$G^A = G^{FS}R - G^{os}R \qquad (33)$$
$$G^F = G^A + G^o \qquad (34)$$
$$G^o = G^F - G^A \qquad (35)$$
$$G^T = Q \times G^{ACF} \qquad (36)$$
$$G^T = W \times G^{AP} \qquad (37)$$
$$G^{ACF} = \frac{G^T}{Q} \qquad (38)$$
$$Q = \frac{G^T}{G^{ACF}} \qquad (39)$$
$$G^{AP} = \frac{G^T}{W} \qquad (40)$$
$$W = \frac{G^T}{G^A} \qquad (41)$$

where G^A = grains of moisture added per pound or cubic foot of air passing through humidifier

G^{FS} = saturated vapor condition of air at temperature during humidification. This is expressed as grains per pound per cubic foot of air.

R = relative humidity of air at temperature during humidification application

G^{os} = saturated vapor condition of air at temperature of air that is entering heating system

G^F = grains of moisture vapor per pound or per cubic foot of air leaving air washer

G^o = grains of moisture vapor per pound or per cubic foot of air entering air washer

G^T = total grains per hour of moisture vapor added to air in humidifier for maintaining proper humidity in interior spaces

Q = cubic feet of air per hour passing through humidifier

G^{ACF} = grains of moisture vapor added to each cubic foot of air passing through humidifier

W = pounds of air per hour passing through humidifier

G^{AP} = grains of moisture vapor added to each pound of air passing through humidifier

Relative Humidity. Formulas for calculating relative humidity and grains of moisture vapor per pound or cubic foot of air follow.

$$R = \frac{G^p}{G^s} \qquad (42)$$
$$G^p = R \times G^s \qquad (43)$$
$$G^s = \frac{G^p}{R} \qquad (44)$$

where R = the relative humidity of the air and moisture vapor mixture expressed in per cent

G^p = the grains contained in a partial saturation of the air and moisture vapor mixture per pound or cubic foot of mixture

G^s = the grains contained in a saturated mixture of air and moisture vapor, per pound or cubic foot of mixture

Density. Formulas for calculating density or weight per cubic foot of air and cubic feet per one pound of air are as follows:

$$CF = \frac{1}{D} \tag{45}$$

$$D = \frac{1}{CF} \tag{46}$$

where C = cubic feet of air

1 = number one

D = density or weight per cubic foot of air

Grains of Moisture. The formula for calculating grains of moisture vapor in a mixture of air and vapor is:

$$G^p = \frac{W^E \times G^E + W^1 \times G^1}{W^E + W^1} \tag{47}$$

where G^p = grains of moisture vapor mixture of air from exterior and interior spaces per pound of air

W^E = pounds of dry air from exterior spaces

G^E = grains per pound of dry air from exterior spaces

W^1 = pounds of dry air from interior or recirculated air

G^1 = grains per pound of dry air from interior or recirculated air

Temperature of Air Mixture. The formula for calculating the temperature of a mixture of exterior and interior or recirculated air is:

$$T^M = \frac{T^E \times P^E + T^1 \times P^1}{100} \tag{48}$$

where T^M = temperature of air entering heating system

T^E = temperature of air entering heating system from exterior spaces

P^E = percentage of total air supply passing from exterior spaces to heating system

T^1 = temperature of air entering heating system from interior spaces or recirculated air

P^1 = percentage of total air supply passing from interior spaces or recirculated air

100 = 100%

Density Variation. The formula for calculating density variation or change due to the temperature gain or loss is:

$$D^N = D^o \times \frac{T + t^\iota}{T + t^\wedge} \quad \begin{array}{l} \text{For an increase in the air temperature} \\ \text{or} \\ \text{For a decrease in the air temperature} \end{array} \tag{49}$$

where D^N = new density or weight of air per cubic foot due to temperature variation

D^o = original density or weight of air per cubic foot due to original temperature

t^o = original temperature of air

t^N = new temperature of air

T = absolute temperature of air

Heat Units. Formulas for calculating heat units (B.t.u.) required for providing moisture vapor to or removing it from air for air conditioning are:

$$H^P = \text{B.t.u.'s } G^F - \text{B.t.u.'s } G^o \tag{50}$$

$$H^T = W^A \times \text{B.t.u.'s } G^F - \text{B.t.u.'s } G^o \tag{51}$$

$$H^T = W^A \times H^{PA} \tag{52}$$

$$H^T = W^w \times H^{pw} \tag{53}$$

where H^P = B.t.u.'s or heat units for providing quantity of moisture vapor for application to air by humidification

G^F = grains of moisture vapor per pound of air for humidification of air for interior spaces

G^o = grains of moisture vapor per pound of air entering heating system

B.t.u. = heat units of the expressed grains of moisture vapor

H^T = total heat units necessary for moisture vapor application for humidification of air

W^A = pounds of air per hour passing through humidifiers

H^{PA} = heat units for moisture vapor per pound of air entering interior spaces from humidifier

W^W = pounds of moisture vapor for humidification application

H^{PW} = heat units required per pound of moisture vapor for humidification of air

Grains of Moisture Vapor. The following formulas are used in calculating grains of moisture vapor per pound or per cubic foot of air passing from humidifier:

$$G^{PA} = \frac{G^T}{W^A} \tag{54}$$

$$G^C = \frac{G^T}{CF^A} \tag{55}$$

where G^{PA} = grains of moisture vapor per pound of air passing from humidifier

G^T = total grains of moisture vapor passing from humidifier

W^A = pounds of air per hour passing through humidifier

G^{CFA} = grains of moisture per cubic foot of air passing from humidifier

CF^A = cubic feet of air per hour passing through humidifier

Temperature of Water. The formula for calculating the temperature of water that is required for use in the air washer for humidification of air for heating application is:

$$T^E = \frac{H^T}{W} + T^F \tag{56}$$

The formula for calculating temperature of water considering the various losses of the air washer as a heat transfer equipment or efficiency of the air washer unit is:

$$T^E = \frac{H^T}{W E^F} + T^F \tag{57}$$

The formula for calculating the temperature rise of the water of the air washer above the initial water or air temperature leaving the air washer for humidification, is:

$$T^L = \frac{H^T}{W} \text{ and } T^L \quad \frac{H^T}{WE^F} \tag{58}$$

The formula for calculating the pounds of water required for humidification application in the air washer in use as a humidifier is:

$$W = \frac{H^T}{T^L} \tag{59}$$

where T^E = temperature of water for air washer in use for humidification of air

H^T = total heat units necessary for humidification in air washer

W = pounds of water for air washer per hour

T^F = leaving or final temperature of water following contact with air in air washer

E^F = over-all efficiency of air washer as heat transfer equipment for humidification application

T^L = temperature loss of water or difference in temperature between inlet water and leaving water of air washer

The formula for calculating the temperature of the water that enters the air washer for dehumidification and cooling is:

$$T^E = T^L = T^D \tag{60}$$

where T^E = temperature of water entering air washer

T^L = temperature of water leaving air washer

T^D = temperature drop or difference between temperature of water entering air washer and that leaving air washer

Heat Units of Air and Vapor Mixture. The formula for calculating heat units of air and vapor mixture with mixing of exterior and recirculated air passing to heating system is:

$$H^M = (H^{DAE} + R \times H^L) \times P + (H^{DAR} + R \times H^L) \times P \tag{61}$$

where H^M = heat units of exterior and recirculation air passing to heating system

H^{DAE} = heat units per pound of dry air from exterior spaces

R = relative humidity expressed as a decimal, for exterior and interior spaces

H^L = latent heat units required for moisture vapor of exterior and interior or recirculated air of air mixture

P = per cent of total air required for exterior and recirculation for air mixture passing to heating system

H^{DAR} = heat units per pound of dry air from interior spaces or recirculated air from interior spaces

Pounds of Moisture Vapor. The following formulas are used for calculating pounds or fractional pounds of moisture vapor of air:

$$W^P = \frac{G^{AP}}{7000} \tag{62}$$

$$W^P = \frac{G^{CFA}}{7000} \tag{63}$$

where W^P = pounds or fractional pounds of moisture vapor

G^{PA} = grains of moisture vapor per pound of air

7,000 = grains per pound

G^{CFA} = grains of moisture vapor per cubic foot of air

Weight of Air. The formula for calculating the weight of air that is required per hour for humidification is:

$$W^A = \frac{G^T}{G^A} \tag{64}$$

where W^A = weight of air that is required per hour for the medium as a carrier for necessary moisture vapor for humidification application

G^T = total grains of moisture vapor necessary for humidification of air for heating application

G^A = additional grains per pound of air passing in heating system for proper humidification

Pounds of Moisture Vapor. The formula for calculating pounds of moisture vapor required for humidification or dehumidification of air is:

$$P^M = \frac{G^T}{7000} \tag{65}$$

where P^M = pounds of moisture vapor for humidification or dehumidification of air for interior spaces

G^T = total grains of moisture vapor for humidification or dehumidification of air for interior spaces

7,000 = grains per pound

Total Grains of Moisture Vapor. The formula for calculating total grains of moisture vapor for humidification of the air is:

$$G^T = G^{TA} + G^{TO} \tag{66}$$

where G^T = total grains of moisture vapor

G^{TA} = total grains of moisture vapor added

G^{TO} = total grains of moisture vapor original

The formula for calculating grains of moisture vapor to be added to the air for humidification is:

$$G^A = G^T - G^I \tag{67}$$

G^A = grains of moisture vapor added to air for humidification

G^T = total grains of moisture vapor required for humidification of air

G^I = total grains of moisture vapor from interior spaces mixed with air

Cubic Feet of Air. The formula for calculating cubic feet of air is:

$$CF^T = W^{PT}CF^P \tag{68}$$

$$W^{PT} = \frac{CF^T}{CF^P} \tag{69}$$

where CF^T = total cubic feet of air

W^{PT} = total weight of air expressed in pounds

CF^P = cubic feet of air per pound

Weight of Air. The formulas for calculating weights of air from exterior and interior spaces are:

$$W^E = W^T P \tag{70}$$

$$W^I = W^T P \tag{71}$$

where W^E = weight of air from exterior spaces

W^T = total weight of air

P = per cent of total amount of air

W^1 = weight of air from interior spaces

Temperature of Interior and Exterior Air Mixture. The formula for calculating the temperature of the mixture of air from interior and exterior spaces is:

$$T^M = \frac{W^E T^E + W^1 T^1}{W^E + W^1} \tag{72}$$

where T^M = temperature of air mixture from interior and exterior spaces

W^E = weight of air from exterior spaces

T^E = temperature of air from exterior spaces

W^1 = weight of air from interior spaces

T^1 = temperature of air from interior spaces

Grains Moisture Vapor to Be Removed. The formula for calculating grains of moisture vapor to be removed from the air for dehumidification is:

$$G^R = (G^E + G^1) - G^A \tag{73}$$

where G^R = total grains of moisture vapor to be removed from air for dehumidification of air

G^E = total grains of moisture vapor from exterior spaces

G^1 = total grains of moisture vapor from interior spaces

G^A = total grains of moisture vapor mixed with air entering interior spaces from dehumidifier

Temperature of Water for Dehumidification. The formulas for calculating temperature of water entering the air washer for dehumidification are:

$$T = T^E + T^L \tag{74}$$
$$T^E = T - T^L \tag{75}$$

where T = temperature of water leaving air washer

T^E = temperature of water entering air washer

T^L = temperature increase of water in air washer

Total Grains of Moisture. Formulas for calculating total grains of moisture are:

$$G^T = G^E + G^1 \tag{76}$$
$$G^T = G^E P + G^1 P \tag{77}$$

where G^T = total grains of moisture vapor

G^E = grains of moisture vapor mixed with air from exterior spaces

P = per cent of interior and exterior air quantity

G^1 = grains of moisture vapor mixed with air from interior spaces

Transfer of Total Heat Units in Dehumidifier. The formula for finding total heat units transfer in the dehumidifier is:

$$H^T = (H^{DAE} + R \times H^L) \times P + (H^{DAR} + R \times H^L) \times P - H^s + H^L \tag{78}$$

where H^T = total heat units transfer in dehumidifier

H^{DAE} = heat units sensible exterior air

R = relative humidity

H^L = heat unit (latent heat)

P = percentage of total air required

H^s = sensible heat of air leaving washer

H^L = latent heat of air leaving washer

H^{DAR} = sensible heat interior air

Tables 76 and 77. General Data (Abbreviated) on Types A and B Buffalo Washers

Size Number	Capacity, Cu. Ft. Air per Minute	Size Make-Up Water Supply (Inches)	Sq. Ft. Spray Chamber Area	Width Over Angles (Inches)	Height Over Angles (Inches)	Length (Inches)	Square Feet of Washing Surface	Number of Spray Nozzles	Number of Flooding Nozzles	Gals. per Minute, Spray Nozzles	Gals. per Minute, Flooding Nozzles	Total Gallons per Minute	Pump Size (Inches)	Hp. Motor 1750 R.p.m.
TYPE A														
1A	1600	¾	3.16	17¼	49½	86⁵⁄₁₆	28.6	5	4	9	4	13	1½	1½
2A	3300	¾	6.07	33	49½	86⁵⁄₁₆	60.5	10	9	18	9	27	1½	1½
3A	5100	¾	10.2	48¾	49½	86⁵⁄₁₆	92.5	15	14	27	14	41	1½	2
4A	6800	¾	13.7	64½	49½	86⁵⁄₁₆	124.	20	20	36	20	56	1½	2
5A	8600	¾	17.1	80	49½	86⁵⁄₁₆	155.	25	25	45	25	70	2	2
6A	10300	¾	20.6	95¾	49½	86⁵⁄₁₆	187.	30	30	54	30	84	2	3
7A	12000	¾	24.1	111½	49½	86⁵⁄₁₆	210.	35	35	63	35	98	2	3
2B	4600	⅞	9.17	33	61½	86⁵⁄₁₆	84.5	12	9	22	9	31	1½	1½
3B	7000	⅞	14.0	48¾	61½	86⁵⁄₁₆	129.	18	14	33	14	47	1½	2
4B	9400	⅞	18.8	64½	61½	86⁵⁄₁₆	174.	24	20	43	20	63	2	2
5B	11700	⅞	23.5	80	61½	86⁵⁄₁₆	218.	30	25	54	25	79	2	3
6B	14200	⅞	28.3	95¾	61½	86⁵⁄₁₆	263.	36	30	65	30	95	2	3
7B	16600	⅞	33.1	111½	61½	86⁵⁄₁₆	308.	42	35	76	35	111	2	3
8B	19000	⅞	38.0	127¼	61½	86⁵⁄₁₆	351.	48	40	87	40	127	2	3
4C	14550	⅞	29.0	64½	85½	86⁵⁄₁₆	273.	40	20	72	20	92	2	3
5C	18200	⅞	36.3	80	85½	86⁵⁄₁₆	343.	50	25	90	25	115	2	5
6C	21900	⅞	43.7	95¾	85½	86⁵⁄₁₆	413.	60	30	108	30	138	2	5
7C	25500	⅞	51.2	111½	85½	86⁵⁄₁₆	483.	70	35	126	35	161	2	5
8C	28800	⅞	58.6	127¼	85½	86⁵⁄₁₆	553.	80	40	144	40	184	2	5
TYPE B														
1A	1600	⅞	3.16	17¼	49½	111½	28.6	10	4	18	4	25	1½	1½
2A	3300	⅞	6.07	33	49½	111½	60.5	20	9	36	9	48	1½	1½
3A	5100	⅞	10.2	48¾	49½	111½	92.5	30	14	54	14	68	1½	2
4A	6800	⅞	13.7	64½	49½	111½	124.	40	20	72	20	92	2	2
5A	8600	⅞	17.1	80	49½	111½	155.	50	25	90	25	115	2	2
6A	10300	⅞	20.6	95¾	49½	111½	187.	60	30	108	30	138	2	3
7A	12000	⅞	24.1	111½	49½	111½	219.	70	35	126	35	161	2	3
2B	4600	⅞	9.17	33	61½	111½	84.5	24	9	41	9	50	1½	3
3B	7000	⅞	14.0	48¾	61½	111½	129.	36	14	66	14	80	2	3
4B	9400	⅞	18.8	64½	61½	111½	174.	48	20	86	20	106	2	3
5B	11700	⅞	23.5	80	61½	111½	218.	60	25	108	25	133	2	3
6B	14200	⅞	28.3	95¾	61½	111½	263.	72	30	130	30	160	2	3
7B	16600	⅞	33.1	111½	61½	111½	308.	84	35	152	35	187	2	5
8B	19000	⅞	38.0	127¼	61½	111½	351.	96	40	171	40	211	2	5
4C	14550	⅞	29.0	64½	85½	111½	273.	80	20	130	20	161	2	5
5C	18200	⅞	36.3	80	85½	111½	343.	100	25	206	25	206	2	5
6C	21900	⅞	43.7	95¾	85½	111½	413.	120	30	237	30	237	2	5
7C	25500	⅞	51.2	111½	85½	85½	483.	140	35	321	35	326	2	5
8C	28800	⅞	58.6	127¼	85½	85½	553.	160	40	360	40	370	4	7½
9C	33000	⅞	66.0	143	85½	85½	623.	180	46	396	46	442	4	7½
10C	38700	⅞	73.4	158½	85½	85½	683.	200	51	576	51	654	4	10
11D	40100	⅞	80.7	174½	85½	85½	763.	220	56	542	52	623	4	15
11D	50200		118.2	253										15
10E	62700		126.2	190					574		61	657	1	15

Selection of Air Washers. Air washers are selected on the basis of air capacity, and standard practice usually assumes a flow velocity of 500 feet per minute through the washer proper. This, of course, will definitely determine the area required and the nearest standard size washer can be selected from a catalogue on this basis. Tables 76 and 77 show typical general data for various sized washers as manufactured by one company. Knowing the requirements of a given job the proper washer can be selected from the tables.

Example 1. It is required to select a Buffalo Air Washer for a school which houses a maximum of 500 pupils. All fresh air is to be used. The air must be heated from 0°F. to 70°F. and delivered at 40% relative humidity. If 30 cubic feet per minute is allowed per pupil, the air requirement will be 15,000 c.f.m. Where washing of air is the first consideration, the A type washer generally is used. If a high degree of cooling or humidification is required, type B should be used. For this problem the A type is selected. Referring to Table 76 either a number 7B or 4C washer might fit the average building. The 4C is selected, for example, which has a capacity of 14,550 c.f.m. This is not quite enough, so the velocity may be increased slightly to obtain the desired capacity of 15,000 c.f.m. Then the frictional resistance of the air through the washer will be increased slightly to obtain the desired capacity. The frictional resistance will thus be higher than the rated .25 inch (see Table 78).

Table 78. Performance Data, Buffalo Air Washers

Type of Air Washer	Air Velocity, Feet per Minute					Cooling Per Cent of Wet-Bulb Depression
	400	450	500	550	600	
	Frictional Resistance—Inches of Water					
A	.16	.20	.25	.31	.38	70
B	.16	.20	.25	.31	.38	90

Note: The balance of this example refers to "Ventilating Systems," Chapter VI, and to such systems as shown in Figs. 41, 42 and 43 in Chapter VI.

Fig. 181 illustrates the exact system which can be used in this example. Heater *A* will be turned on by the thermostat in the fresh air duct, when the outside temperature falls below 35°F., in order to prevent the washer water from freezing. A heater will be selected for this purpose capable of raising the temperature of the air from 0°F. to about 30°F.

Note: Explanation and selection of heating coils is given on page 385.

To have a final temperature of 70°F. and 40% relative humidity, the dew point temperature must be 45°F., as shown by the Psychrometric Chart. Thus the washer must heat the air from 30°F., containing practically no moisture (corresponding to a wet-bulb temperature of about 21°F.), to a condition of saturation at 45°F. This will require $17.7 - 7.5 = 10.2$ B.t.u.'s per pound of dry air, or the difference in total heat. From the Psychrometric Chart we find that

the volume of air at 70°F. and 40% relative humidity is 13.5 cubic feet per pound. Therefore the heat required is $10.2/13.5 = .755$ B.t.u. per cubic foot or $.755 \times 60 \times 15,000 = 680,000$ B.t.u.'s per hour for the air used. In figuring the amount of steam required for a closed heater control, the latent heat of the steam at gauge pressure should be used, while for open heater control the heat of the liquid is also available, so that the difference in the total heat of steam at gauge pressure and at the desired air dew point (45°F. in this example) should be calculated and applied. Thus, if a closed heater control of 5 pounds steam pressure is selected, the steam required for this example is $680,000/960.7 = 707$ pounds per hour. Heater D, shown in Fig. 181, will therefore heat the air leaving the washer from 45°F. saturated to 70°F. and is the correct size of heater to use. Since no moisture is added after the air leaves the washer, it will have a constant dew point of 45°F. and when it reaches the temperature of 70°F. the relative humidity will be 40%, as specified.

Example 2. It is required to select a Buffalo Air Washer to cool a theatre which has a capacity of 2,000 people. If we allow 30 cubic feet of air per minute for each person, the amount of air required is 60,000 c.f.m. For this class of work a type B washer is selected which is capable of lowering the air temperature 90% of the wet-bulb depression. (See Table 78.) The size of washer selected depends upon the nature of the building construction. Thus either a 16C or 12D or 10E type washer (see Table 77) might serve the purpose.

Most of the heat, which must be removed from the theatre, is thrown off by the individuals in attendance and will amount to about 400 B.t.u.'s per hour per person. This will total 800,000 B.t.u.'s per hour or 13,300 B.t.u.'s per minute. If we assume that the outside temperature is 80°F. with a wet-bulb temperature of 65°F. corresponding to 45% relative humidity, we find that the volume of this air is 13.8 cubic feet per pound of dry air present. Then the weight of air handled will be $60,000 \div 13.8 = 4,350$ pounds per minute. Since 4,350 pounds per minute must absorb heat at the rate of 13,300 B.t.u.'s per minute there will be 3.06 B.t.u.'s per pound of dry air added to the total heat of the air. The humidifier will cool the air 90% (see Table 78) of the wet-bulb depression or $.90 (80 - 65) = 13.5°$. That is, the air will leave the washer at a dry-bulb temperature of $80 - 13.5 = 66.5°$F. and at a wet-bulb temperature of 65°F, corresponding to a dew point of 64.3°F. The total heat of this air is 29.5 B.t.u.'s per pound, but when mixed with room air will increase 3.06 B.t.u.'s per pound due to heat from people. Thus the room air will have a total heat of 32.56 B.t.u.'s per pound of dry air, corresponding to a wet-bulb temperature of 69°F.

Of the 400 B.t.u.'s per hour per person approximately 100, or 25%, will be for latent heat* due to expired moisture, so that $.25 \times 3.06 = .765$ B.t.u. per pound of dry air has been added to the moisture content. At this temperature it will require $1055/7000\dagger = .151$ B.t.u. per grain of moisture evaporated. Therefore, $765/.151 = 5.1$ grains of moisture per pound of dry air has been added, corresponding to an increase in the dew point from 64.3°F. to 65.7°F. Having room air now at 69°F. wet bulb and 65.7°F. dew point, the dry-bulb temperature will be found to be 76°F. Actually it will be slightly higher due to heat from electric lights, etc.

Thus it can be seen that the air washer has a cooling effect and that with-

*See page 17.
†See page 321.

out it a considerable rise in temperature would cause discomfort. Any further cooling would require a dehumidifier using either cold water or refrigerant.

Example 3. A large commercial building will require air conditioning for the comfort of the occupants and maintenance of commodity. The air in this building is to be cooled and dehumidified by an air washer.

The interior space of the building is to be maintained at 80°F. with 50% relative humidity. The exterior air is 95°F. with 50% relative humidity. The building requires 40,000 pounds of air per hour for cooling and dehumidification.
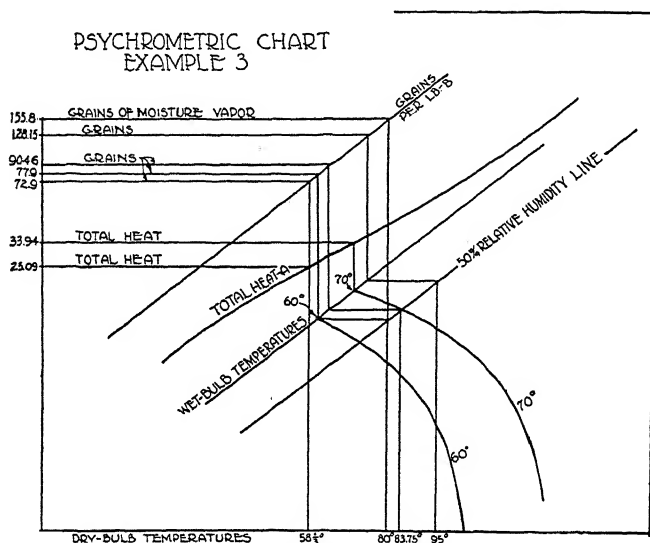
Fig. 182. Abbreviated Psychrometric Chart

Of this amount 25% is supplied from the exterior and 75% is supplied from the interior by recirculation. The recirculated air from the interior enters the air-conditioning system at 80°F. and 50% relative humidity . There are 50 occupants who are very active, which causes the elimination of 1,200 grains of moisture vapor per hour for each. The servicing equipment provides 140,000 grains of moisture vapor per hour through the air of the interior.

The cooling and dehumidification of the air for this building is to be provided by heat transfer from the air to water in an air washer. The moisture vapor and the heat transfer can be obtained by using the Psychrometric Chart or Table 75.

Note: Fig. 182 is an abbreviated Psychrometric Chart of the type shown in the back of the book. In Fig. 182 only the lines essential to this example are shown. The chart is used exactly as explained on page 12, Chapter II.

Example 3A. What is the quantity of air from the exterior and interior spaces in pounds per hour?

Example 3B. What is the temperature of the mixture of the exterior and interior air, passing through the air-conditioning system?

Example 3C. How many grains of moisture vapor per pound of air enter the air washer?

Example 3D. What is the heat transfer, expressed in heat units or B.t.u.'s for the cooling and dehumidification of the air for the interior?

Example 3E. What are the weight and the entering temperature of the water for use in the air washer for cooling and dehumidification of the air for the interior?

Example 3F. Explain why cooling of the air is necessary for the dehumidification of the air.

Solution 3A. The 40,000 pounds of air per hour multiplied by 25%, quantity of total air supply from exterior, equals 10,000 pounds—which is the amount of exterior air required. In like manner, multiplying 40,000 by 75% equals 30,000 pounds of interior air required.

Formulas (37) and (38) are used for the above calculations.

Solution 3B. The 95°F. temperature of exterior air multiplied by 25%, percentage of air from the exterior, plus the 80°F. temperature multiplied by 75%, percentage of air from the interior, and divided by 100 equals 83.75°F. This is the temperature of the mixture of exterior and interior air. The same results may be obtained by using Formula (16).

Solution 3C. Either Table 75 or the Psychrometric Chart, Fig. 182, can be used to determine the grains of moisture vapor for 80°F. and 50% relative humidity. Fig. 182 shows that 80°F. air saturated contains 155.8 grains per pound. This is shown also in Table 75. At 50% relative humidity 80°F. air contains 77.9 grains per pound. This is shown in Fig. 182 and could be calculated by multiplying the table value of 155.8 by 50%. Air at 95°F. and relative humidity of 50% contains 128.15 grains of moisture vapor per pound. This is calculated as explained for the 80°F. air. The 77.9 grains multiplied by 75% plus the 128.15 grains multiplied by 25% equals 90.4625 grains of moisture vapor per pound of air mixture entering the air washer.

This example can be solved directly by Formula (15).

Note: In substituting for G^E and G^I, in Formula (15) the reader is cautioned to use the moisture value of the air without taking the percentage of the amounts of interior and exterior air into consideration.

Solution 3D. To determine the heat transfer for cooling and dehumidifying, Table 75 or the Psychrometric Chart, Fig. 182, may be used.

The air of the interior at 80°F. and 50% relative humidity will have 77.9 grains of moisture vapor per pound. The air mixture entering the air washer at 83.75°F. will have 90.4625 grains of moisture vapor per pound. Then 90.4625 multiplied by 40,000, plus 60,000 grains (1200 multiplied by 50 people) plus 140,000 grains (servicing equipment) equals 3,818,500 grains of moisture vapor per hour. The 77.9 grains multiplied by 40,000 equals 3,116,000 grains of moisture vapor per hour allowed in the interior. Subtracting 3,116,000 from 3,818,500 equals 702,500 grains of moisture vapor per hour to be removed by the 40,000 pounds of air per hour for the interior.

The 702,500 grains divided by 40,000 equals 17.5625 grains of moisture vapor per hour removed per pound of air to provide 50% relative humidity with 80°F. temperature for the interior. The air leaving the air washer will have 90.4625 minus 17.5625 = 72.9 grains of moisture vapor with saturation or wet-bulb temperature of $58\frac{1}{3}$°F.

The air and moisture vapor mixture of 58⅓°F., wet-bulb temperature, has a total heat of 25.093 B.t.u.'s per pound of the mixture that is leaving the air washer. (See Table 75 and Fig. 182.) At 83.75 degrees the air has a total heat of 33.94 B.t.u.'s per pound. The transfer of heat in B.t.u.'s, due to the heat exchange, is the difference between 33.94 and 25.093 which equals 8.847 B.t.u.'s per pound of air. The 8.847 multiplied by 40,000 equals 353,880 B.t.u.'s or the total heat transfer of the water and the air passing through the air washer per hour. Formula (78) can be used directly in arriving at the same answer.

To determine the actual heat quantity required, the efficiency of the air washer as a heat transfer apparatus must be considered. We assume a washer efficiency of 75%. Then 353,880 B.t.u.'s divided by 75% equals 471,840 B.t.u.'s per hour, which is the heat exchange for cooling and dehumidifying the air. This quantity allows for the various losses of the air washer during operating periods.

Solution 3E. The air washer requires from 4 to 8 gallons of water per minute for cleaning, cooling, and dehumidification of 1,000 cubic feet of air per minute. For this example 8 gallons per hour will be assumed.

The air entering the air washer at 83.75°F. will have about 13.923 cubic feet of air per pound. This is not shown in Fig. 182 because it is easier to take it from Table 75. The 13.923 cubic feet multiplied by 40,000 pounds equals 556,920 cubic feet of air per hour or 9,282 cubic feet per minute. This means that there are 9.282 thousands of air per minute. Then 9.282 multiplied by 8 equals 74.256 and this multiplied by 8.34 (pounds per gallon) equals 619.295 pounds of water per minute for the air washer. Then 619.295 multiplied by 60 equals 37,157.7 pounds of water per hour.

To determine the temperature of water for the air washer, the heat transfer expressed in B.t.u.'s per hour is divided by the weight of water per hour which equals the temperature gained as the air passes through the air washer. The 353,880 B.t.u.'s divided by 37,157.7 equals 9.521°F. temperature increase of the water in the air washer. As the temperature of water leaving the air washer is the same as that of the air leaving the air washer, (58⅓°F.), the temperature of the water entering the air washer is 58.333°F. minus 9.521°F. or 48.812°F. To determine the actual temperature difference of the water entering and leaving the air washer for cooling and dehumidification, the efficiency of the air washer as a heat exchange apparatus must be considered. The usual average thermal efficiency of the air washer is about 75%. In calculating the actual temperature difference of the water entering and leaving the air washer, the 353,880 is divided by 75% which equals 471,840 B.t.u.'s per hour required for the cooling and dehumidifying operation. Then, when considering the efficiency, it is necessary to divide the 471,840 by 37,157.7 which equals 12.69°F. the temperature increase of the water passing through the air washer. The water leaving the air washer is at the same temperature as the air passing from the washer, as previously explained, or 58⅓°F. Then 58.33 − 12.69 equals 45.64°F. the temperature of water entering the air washer. Formula (75) can be used for the above example.

Solution 3F. The cooling of the air, for dehumidification, is necessary in order to remove the moisture vapor by lowering the air temperature to and below the dew point, which causes condensation of the air-borne moisture. The heat transfer or exchange that lowers the temperature of the air is sensible heat whereas the heat transfer that causes condensation, or dew formation, is latent heat. The sensible and latent heat combined compose what is called "total heat of air."

PRACTICE PROBLEMS

1. An industrial building requires 200,000 pounds of air per hour for heating and humidification. Of the total air required for the heating and ventilating system, 40% is from the exterior at a temperature of 35°F. and 75% relative humidity and 60% is from the interior (recirculated air), at a temperature of 62°F. and 40% relative humidity. The interior of the building is to be maintained at 65°F. and 45% relative humidity. There are 1,900,000 grains of moisture vapor provided by the occupants and equipment of the interior. The preheating radiation principle and the air washer are used for heating and humidification.

1A. What are the weights of air from the exterior and interior spaces?

1B. What is the temperature of the mixture of the air from the exterior and interior passing to the heating and ventilating system?

1C. What is the heat transfer quantity expressed in B.t.u.'s per hour for the heating and humidification of the air?

1D. What is the quantity of moisture vapor for the humidification of the air for the interior?

1E. Determine the weight of water necessary for the air washer if the entering water for humidification is 85°F. and the water leaving the air washer is 65°F. Calculate the weight per hour.

2. A large mercantile and office building requires 400,000 cubic feet of air per hour from the exterior for heating and ventilating of the interior. The interior is to be maintained at 70°F. and 45% relative humidity. There are 200 occupants of the building. Each occupant will provide 700 grains of moisture vapor per hour for mixing with the air. The air of the exterior is 30°F. and 60% relative humidity.

2A. What method of humidification would you use for the requirements of the building?

2B. What is the air temperature for distribution through the air washer to provide 45% relative humidity for the interior spaces?

2C. What is the total heat transfer in B.t.u.'s for humidification when using water from the air washer for heating and moisture vapor application?

2D. What is the temperature of the water in the air washer for the heating and humidification of the air, that is, considering the efficiency of the air washer as a heat transfer apparatus? Assume 90% efficiency.

2E. How many pounds of water per hour are required for humidification of the air?

Summary. Air washers as used in the general practice today can be considered primarily as a heat exchange apparatus, and secondarily, as a cleaning equipment. An air washer is logical equipment to serve both purposes. The finely divided water spray exposes a large surface to the air passing through and this, combined with the fact that water has a high coefficient of conductivity, produces excellent conditions for heat transfer.

The problem is usually presented by stating the quantity of air to be handled, giving its initial temperature (both dry bulb and wet

bulb) with the requirement for final temperature, usually the dew point, and also by stating the water temperature available.   This resolves itself into a matter of computation of heat balances and thus a determination of final leaving temperature of water (which often is assumed) and the quantity of water needed to obtain a rational heat exchange.

Evaporative Condensers.   An evaporative condenser is a device designed to replace the air- or water-cooled condenser normally supplied with a refrigerating unit.   As its name implies, the evaporative condenser utilizes the principle of evaporation to remove the heat of condensation from the refrigerant gas.   While the principle of evaporative cooling is not new, its application in the evaporative condenser is of recent origin.

Fans draw or blow air over the condenser which consists of a coil over which cool water is sprayed to condense the refrigerant gas within.   Centrifugal or propeller type fans are used.   These fans are designed to handle large volumes of air at high velocity over the condensing coil and through the water spray.   The water evaporated by the condensing refrigerant is picked up by the air stream and discharged outdoors.

Economies and Advantages of Evaporative Condensers.   Evaporative condensers can be used profitably on any air-conditioning installation where any of the following conditions exist:

1. High water costs. During the past few years electric power rates through the country have been decreasing while water rates, in general, have been increasing. Inasmuch as water is a commodity controlled and distributed by municipalities, there seems to be no likelihood that this trend toward higher water rates will cease.

2. High water temperature. High water temperatures mean large quantities and high condensing temperatures. Both result in high operating costs of water-cooled equipment.

3. High power costs. If power costs are relatively high—three to seven cents per kilowatt hour—the savings are made with an evaporative condenser are considerable.

4. Low maximum wet-bulb temperature. This increases the capacity of an evaporative condenser and results in greater operating economy if average wet-bulb temperature is also low.

5. Inadequate water supply or disposal systems. These include the following: (a) low water supply pressure; (b) lack of uniform supply pressure; and (c) lack of adequate disposal systems. Inadequate water supply or disposal systems often cause trouble in summer months when the peak of the water load exists in practically all cities. They often represent difficult problems in water-cooled installations.
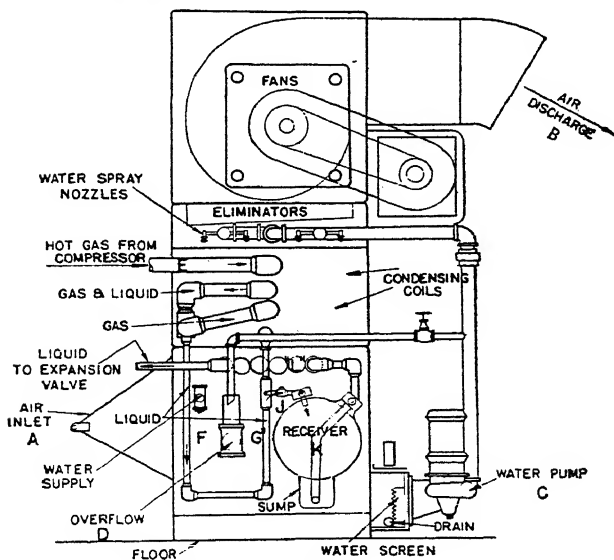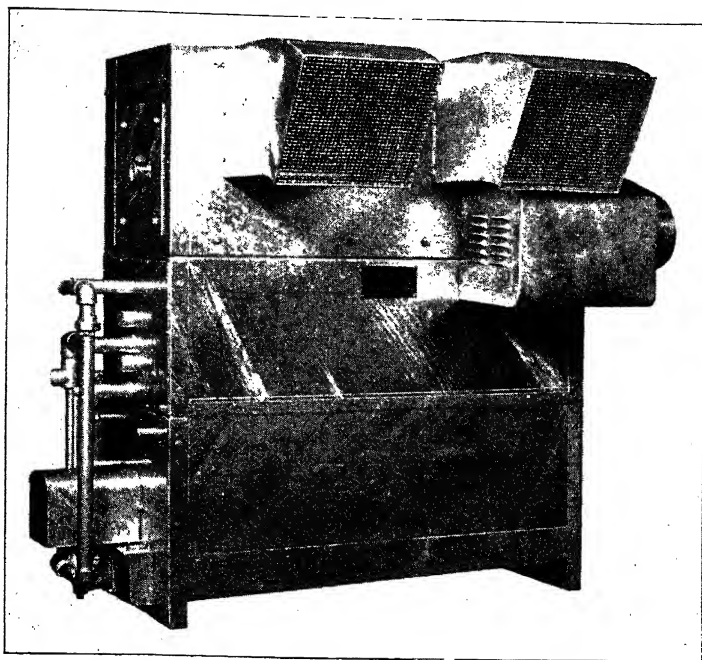
Fig. 183. Carrier Type 9Q Evaporative Condenser

Evaporative condensers have the following advantages:

1. Low compressor power and great compressor capacity because (a) they provide a lower condensing temperature for the same outdoor wet bulb, and (b) incorporate a liquid sub-cooling coil which adds to the efficiency of the system.

2. Low installed cost of system in small plants because of (a) elimination of water-cooled condenser; (b) reduction of compressor size; (c) less cost for piping and pumps; (d) greater freedom in choice of location; and (e) less expensive foundations and supports.

3. Lower fan power than an indoor cooling tower or force draft tower.

4. Low pump power.

5. Wide choice of location because of (a) smaller, more compact construction; (b) less weight; and (c) enclosed self-contained design.

6. High salvage value provided by unitary construction in portable elements.

7. Completely self-contained and weatherproof and may be installed either within a building or outdoors, without additional housing.

Types of Evaporative Condensers. Figs. 183 and 184 show two types of Carrier evaporative coolers in cross section and exterior so their major parts may be studied. Type 9Q, Fig. 183, stands upright on the floor whereas type 9P, Fig. 184, generally is suspended.

Operating Cycle—Type 9Q, Fig. 183. Air enters the unit through the intake screen at A and is drawn up through the spray chamber condensing coils and eliminators, and discharged out of doors at B through outlets or duct work. The air, in passing through the spray chamber and over the condensing coils, removes heat from the refrigerant by evaporation of a portion of the spray water.

The circulating pump C draws water from the spray tank and discharges it above the condensing coils through a series of spray nozzles. The spray nozzles are arranged to distribute the water uniformly over the entire spray chamber. Evaporative cooling, resulting from the air passing over the wet surfaces, condenses the refrigerant in the coil and vaporizes a portion of the spray water. A large excess of spray water is circulated by the pump to keep the surfaces thoroughly wet and to maintain a high saturation efficiency. This large water circulation also washes the coil surfaces clean and maintains the rate of heat transfer.

Make-up water is supplied to the tank through a solenoid valve wired in parallel with the fan motor. This make-up water is necessary for replacing the water evaporated, and to provide an overflow for the removal of scum, hardness, and acidity. It prevents scaling and serious corrosion of the condenser surface. The water overflows at D.

The hot gas discharge from the compressor enters the condenser at E. Part of the gas is condensed as it flows through the finned tubes of the condensing coil. This condensed refrigerant is taken off by bleeder connection at F, and the remaining gas fed to the inlet of the next condensing coil. The efficiency of each coil is increased by use of the bleeder connections which remove the liquid and supply only gaseous refrigerant to each condensing circuit. The liquid refrigerant from bleeder lines F and G flows through the check valve J to the receiver K. From the sump in the bottom of the receiver the liquid refrigerant flows through the shut-off valve and liquid sub-cooling coil L to the liquid line and expansion valve. The liquid sub-cooling coil is located in the spray chamber, in the path

of the incoming air and thus removes additional heat from the refrigerant. This removal of heat increases the refrigeration capacity of the compressor and results in more efficient operation of the system.

*Specifications—Type 9Q Evaporative Condensers

The following Carrier specifications for type 9Q explain in detail construction, parts, connections, etc. Table 79 gives ratings.
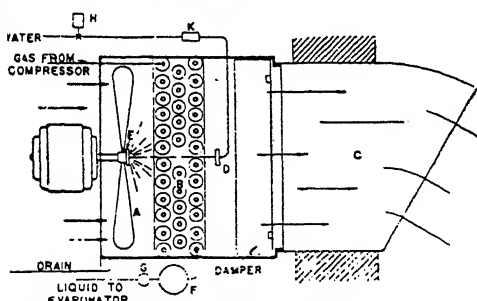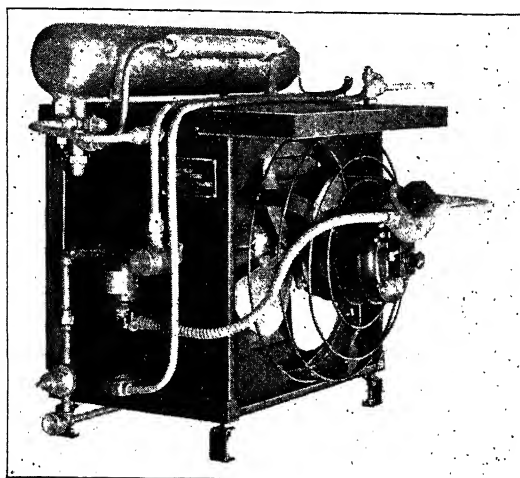
Fig. 184. Carrier Type 9P Evaporative Condenser

Standard Evaporative Condensers include: **Fan Section** with fans, motors, belts, pulleys, and outlets: **Spray and Coil Section,** with sprays, eliminators, and finned type condensing coil; **Drip Pan Section,** with pump and motor, water control assembly, and air intake trough. All units are dehydrated and sealed without refrigerant charge.

Casing. The fan, spray, coil, and drip pan section are of pressed steel and welded construction. Each section is hot-dipped galvanized. The unit is weather-

*Carrier Engineering Corp.

proof and may be installed outdoors. Removable doors are provided for access to internal elements.

Fans, Motor, and Drive. Quiet, high efficiency, multi-blade centrifugal fans. Standard motors are 1,725 r.p.m., 220 volt, 3 phase, .60 cycle, placed on special mounting which is effective in sound and vibration absorption and permits easy adjustment of belts. V-belt drive protected by removable ventilated belt guard.

Fan Section. Standard fan section assembled for side discharge. Sizes 9Q2, 9Q6 and 9Q7 may be assembled for top discharge. 9Q9 cannot be assembled for top discharge. Standard outlets are furnished with unit. The entire back of the fan section is removable for access to, and inspection of, the interior of the unit.

Fan Bearings. Ball bearings mounted in adjustable housings. Furnished with grease cup lubrication. The bearing housings are mounted in end plates which may be moved for adjusting the fan clearance. The fan shaft can be adjusted axially or transversely. All adjustments of fan shaft and bearings can be made from the outside of the unit. The complete fan shaft can be removed from either end of the unit.

Fan and Pump Motor Covers. Copper bearing steel, hot-dipped galvanized. The motor covers are removable for access to the motors.

Water Eliminator. Single bend 24-gauge copper bearing galvanized steel, painted one coat black Rubalt paint.

Sprays. Spray nozzles are the centrifugal type, having a $\frac{1}{4}$-inch diameter orifice. The spray nozzle assembly may be removed through the end of the unit.

Pump and Motor. Low pressure, high efficiency, enclosed, non-overloading centrifugal pump, bronze body. Pump motor, same electrical characteristics as fan motor unless otherwise specified.

Water Control Assembly. Includes assembly of shut-off valve, strainer, pressure regulating valve, gauge, solenoid valve, and unions ready for installation in the water supply line. The water control assembly regulates the water supplied to the condenser.

Overflow Connection. 1 pipe connection in the body of the pump.

Drip Pan Section. Supports the fan and coil section and provides a tank for the water. Furnished with an air intake trough which may be mounted on either side of the section.

Condensing Coils. (Methyl chloride or Freon) constructed of Aerofin seamless copper tubing with high efficiency helically wound copper fins. Coil consists of two circuits used separately in two-circuit unit and connected in series in single-circuit unit.

Condensing Coils. (Ammonia.) Steel construction with plate type steel fins. Entire coil assembly is hot-dipped galvanized. Single-circuit coils only available.

Liquid Sub=Cooling Coil. Constructed of Aerofin seamless copper tubing with high efficiency helically wound copper fins. Coil consists of two circuits used separately in two-circuit unit and connected in parallel in single-circuit unit. (Available only on methyl chloride and Freon units.)

Liquid Receiver. Shell is of drawn steel thimbles welded and separated by a division plate at the center forming two receiver chambers for two-circuit unit. Chambers are interconnected for single-circuit unit. Double shut-off valves at inlet and outlet connections. Liquid level test cock located in receiver indi-

cates minimum operating level. A fusible plug is located in the receiver. (Receiver is not furnished for ammonia model.)

Liquid Line Strainer. Constructed of copper tubing and equipped with Mueller Streamline fittings. One strainer for each circuit is shipped with unit for erection in the field. (Not furnished with ammonia model.)

Electrical Connections. Conduit box provided on the motors. No starting switches are provided as standard with the unit.

Water Supply and Drain Connection. On standard units, 1 drain pipe connection at the bottom of the pump, also 1¼-inch diameter overflow located at same level; ½-inch pipe connection also above waterline in tank for water supply.

Intake Damper Boss. Special, for winter operation to maintain head pressure; equipped with dampers to throttle air supply. Methyl chloride and Freon units, equipped with pressure operated damper motor. Ammonia units equipped with pressurestat operating electric damper motor.

Table 79. *Ratings for Type 9Q, Fig. 183

Type	Basic Physical Dimensions			Approx. Refrig. Capacity Tons	Fans							Water Consump. of Pump g.p.m.
	Length In.	Width In.	Height In.		No.	Diam. In.	Standard c.f.m.	Methyl Chloride or Freon Unit		Ammonia Unit		
								r.p.m.	Hp.	r.p.m.	Hp.	
9Q2	47	21	63	10	2	11¼	2000	1530	1	1320	¾	0.6
9Q6	60	28	77	20	2	16	4000	1020	1½	880	1	1.1
9Q7	85	28	77	30	3	16	6000	1020	2	880	1½	1.7
9Q9	85	37	77	40	3	16	8000	1100	3	945	2	2.1

*Courtesy of Carrier Engineering Corp. Ratings for other types can be obtained from the manufacturer.

Filters. With the introduction of air conditioning into all types of structures, the demand for clean air became pronounced. As a result a variety of filtering devices have been created which aim at the purification of air. Dust and dirt are now universally recognized as among the great menaces to health and to industrial processes.

A normal person breathes approximately 17 times per minute. The air sucked into the lungs may contain large quantities of dust, soot, germs, and other matter not conducive to a healthful condition of the body. The nose functions to prevent dust and other particles from gaining entrance into the lungs. Notwithstanding this fact, some of this matter does enter the lungs and bad results sometimes follow—especially if a person's health is not up to normal. Government figures show that each inch of ordinary city air contains over 110,000 particles of dust, and that each particle may contain over

125,000 germs of various kinds. In or near industrial centers like
Gary, Indiana, Pittsburgh, Pa., etc., the dust content per square
inch of air becomes more dense and thus more harmful.

Dust in such quantities is not only a threat to health but also
an annoyance in homes, stores, restaurants, and all other places
where cleanliness is of importance. When it is known that the dust
precipitation in such representative cities as St. Louis, Mo., Cin-
cinnati, Ohio, and Pittsburgh, Pa., is between 85,000 and 255,000
pounds per square mile it is easy to understand how residences, for
example, become "dusty" soon after being cleaned. In industry, dust
is a detriment and may produce heavy financial losses unless properly
controlled.

Thus the filtration of air for residence and industrial uses be-
comes a matter of real importance and one that must be considered
in any system where air conditioning is contemplated. Filters, there-
fore, are a means of removing dust, soot, etc., from the air prior to its
use in residences, offices, and industrial buildings.*

Filters are of three main types, namely: dry, viscous, and
self-cleaning.

Dry Filters. Dry filters are made in standard sizes and rated as
to capacity and resistance to the passage of air. For proper filtration,
ample filter area must be included in the design of the installation so
the air velocity passing through the filters is not excessive. The
proper area is secured by installing the proper number of filters in
an iron or steel frame in the path of the air.

Dry filters may be made so they can be cleaned at regular inter-
vals or provided with inexpensive filter medium which can be replaced
when filled with dirt. This type of filter is sometimes known as the
"air mat." The filtering medium is generally a composition paper
through which the air passes. This type of filter is best suited for
general ventilation, is usually designed for air velocities of about 45
feet per minute, and has a resistance of from .06 to .18 inches of water.
The application of an air mat filter is shown in Fig. 185. Where such
filters are used in industrial applications they have a resistance less
than .3 inch of water and velocities up to 20 feet per minute. These
filters are not recommended in cases where the air is heavily laden

*Air washers serve to clean the air, too, as noted in the part of this chapter devoted to that
apparatus.

with atmospheric dust. If the air mat is used as a dust collector or if the dust concentration is too great, the filter will clog and increase the resistance.
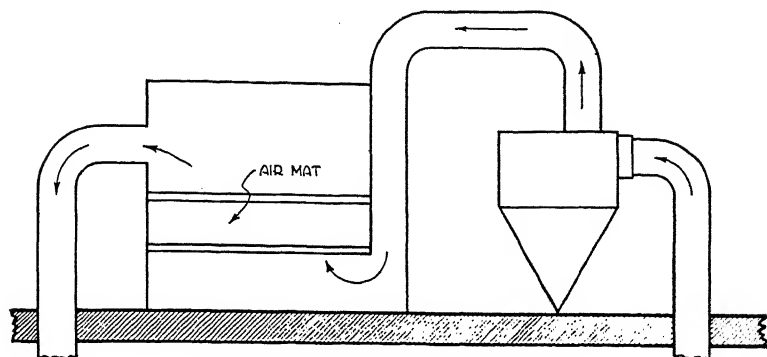
Fig. 185. Application of an Air Mat Dry Filter

Table 80. Group Chart (See page 339.)

Total Height	No. of Units	Capacity and Group Number (in c.f.m.)							
11' 2"	12	7200 1-12	14400 2-12	21600 3-12	28800 4-12	36000 5-12	43200 6-12	50400 7-12	57600 8-12
10' 3"	11	6600 1-11	13200 2-11	19800 3-11	26400 4-11	33000 5-11	39600 6-11	46200 7-11	52800 8-11
9' 4"	10	6000 1-10	12000 2-10	18000 3-10	24000 4-10	30000 5-10	36000 6-10	42000 7-10	48000 8-10
8' 5"	9	5400 19	10800 29	16200 39	21600 49	27000 59	32400 69	37800 79	43200 89
7' 6"	8	4800 18	9600 28	14400 38	19200 48	24000 58	28800 68	33600 78	38400 88
6' 7"	7	4200 17	8400 27	12600 37	16800 47	21000 57	25200 67	29400 77	33600 87
5' 8"	6	3600 16	7200 26	10800 36	14440 46	18000 56	21600 66	25200 76	28800 86
4' 9"	5	3000 15	6000 25	9000 35	12000 45	15000 55	18000 65	21000 75	24000 85
3' 10"	4	2400 14	4800 24	7200 34	9600 44	12000 54	14400 64	16800 74	19200 84
2' 11"	3	1800 13	3600 23	5400 33	7200 43	9000 53	10800 63	12600 73	14400 83
2' 0"	2	1200 12	2400 22	3600 32	4800 42	6000 52	7200 62	8400 72	9600 82
1' 1"	1	600 11	1200 21	1800 31	2400 41	3000 51	3600 61	4200 71	4800 81
No. Units in Width..........		1	2	3	4	5	6	7	8
Total Width.......		1'11"	3'10"	5'9"	7'8"	9'7"	11'6"	13'5"	15'4"

Fig. 186 shows a dry filter in which the filtering medium can be economically replaced. In this filter the filtering action is that of a positive strainer which retains the smallest dust particles in the air passing through it.
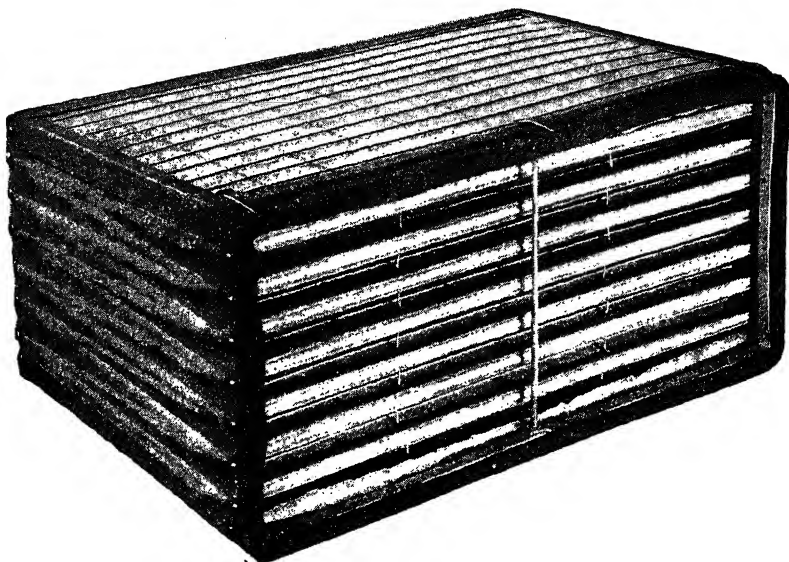
Fig. 186.   Kompak Dry Air Filter with Renewable Medium
Courtesy of Independent Filter Company, Chicago

In using fabrics as the filtering medium, the first problem has always been that of incorporating, within conventional space limitations, sufficient surface to permit of low velocity through the material. No fabric known will function satisfactorily as an air cleaner with high velocity air flow. The higher the velocity, the greater will be the penetration of dust particles into the fibres of the filter material. Conversely, the lower the velocity the less penetration there will be— and the greater the serviceability of the filter medium.

Fig. 187A shows how the air passes through such a filter. Fig. 187B shows a magnified section through a filter medium showing the effect of low velocity air flow (22½ f.p.m.). Loose nature of the deposit (blacker section) does not retard the air flow and by permitting a large accumulation of dust on the filter surface, results in long life of the medium. Fig. 187C, by contrast, shows packing and clogging effect of high velocity (40 f.p.m.) through medium. Dust particles penetrate fibres of fabric and form in a closely packed mat on the surface. The result is reduced dust-holding capacity and the need for frequent replacement.

Kompak Filters. This type of filter may be used in large or small

groups to satisfy any need. The groups are supported by vertical frames or sections, each section being one unit in width and the required number in height. When these vertical "ladders" are bolted together at a job, a rigid frame is formed, free from possible distortion.
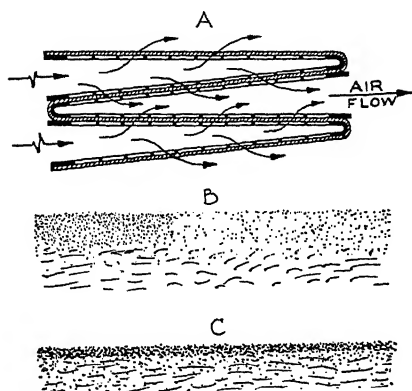
Fig. 187. Detailed Section through Spacer Showing Direction of Air Flow and Effects of Low and High Velocity through Filters

Sheet metal ducts can be attached directly to the outside of the assembled group. See Fig. 188.

The normal rating applied to the Kompak filter is 600 c.f.m. per unit. At this rating, the velocity through the filter material is $22\frac{1}{2}$ f.p.m. and the initial resistance is .15 inches of water. This rating is intended for the average ventilating job where the air is not exceedingly dirty.

Table 80 shows a group chart used in connection with *Kompak filters.* Figures 1 to 12 at the left of the chart indicate the number of units in height. Dimensions indicate total height. Corresponding figures at the bottom indicate the number of units in width, and dimensions show total width. Each unit space shows the group number and capacity (in c.f.m.) of that particular arrangement. The first figure of the group number indicates the number of units in width and the second figure the number in height.

Viscous Filters. The same principle is applied to this type of filter as is used for a common floor oil mop. A coating of viscous film of oil or other adhesive is applied to a series of deflecting surfaces. As air passes through the filter its direction of flow is suddenly

changed. The heavier-than-air dust particles have so much inertia that their direction of flow is not easily changed and they are caught and retained in the viscous oil film.
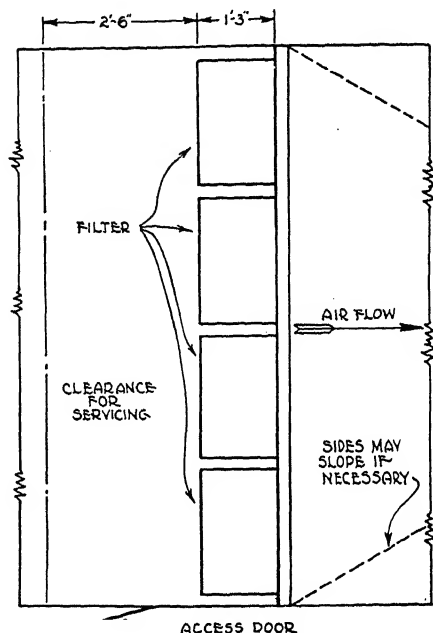
Fig. 188. Installation of Kompak Filters

From the two principles just explained, many filters are made. The following filters (called by trade names) are shown and explained as typical examples of filters used in air-conditioning systems.

Dust-Stop Filter. Fig. 189 shows a section of Dust-Stop filter. This air filter consists of a series of mats of Fiberglas, the parallel faces of which are confined by expanded metal grilles, and bound with fiber-board. The Dust-Stop filter pack is graduated in fiber size from coarse (broom-straw size) at the intake face to fine (the size of human hair) at the discharge face.

The adhesive with which the filter medium is sprayed has extraordinary wetting power, retains its viscosity under operating temperatures ranging from $-15°F$. to $300°F$., will not flow off or charge the air with oil, and has little evaporation. The progressive pack of Fiberglas permits the penetration of fine dust particles into the depth

of the mat and prevents rapid loading of the intake face. Dust-Stop air filters are engineered to provide dust-catching efficiency at low cost of installation and maintenance. They are easy to install and easy to replace. No cleaning, re-oiling, draining, or auxiliary equipment is required. The filter pack, composed of thousands of feet of adhesive-coated glass fibers, presents an exceptionally large dust-catching surface area, yet has low resistance to air flow. The glass fibers, as previously explained, graduate from coarse to fine. They are arranged at random so that the air, in passing through the filter, changes direction many times. This causes dust and dirt particles
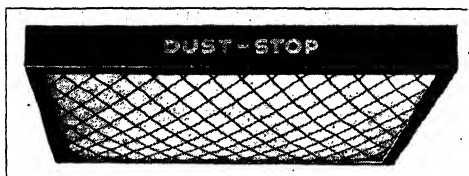
Fig. 189. Dust-Stop Air Filter
Courtesy of Owen-Illinois Glass Company, San Francisco

carried by the air to come in contact with the fibers, and the adhesive holds these particles without stopping the passage of air.

Dust-Stop air filter frame assemblies afford maximum filtering efficiency and are economical to maintain because the replacement filters are low in cost, long of life, and are installed in tandem (face to back). The first, or intake, filters load first, the second filters remaining comparatively clean. Thus only one filter in each unit need be replaced at a time.

For the proper installation of Dust-Stop filters, there have been developed interchangeable, sectional steel frames, which may be combined to meet any c.f.m. requirements. Dust-Stop frames are made in two types, the choice depending upon the space available for installation. Frames are installed on the intake side of the fan.

Dust-Stop Frame Assembly. Fig. 190, shows a Dust-Stop L-type frame assembly, which consists of an assembly of interchangeable L-shaped members of painted steel on which are mounted viscous filters. These frame members may be assembled vertically or horizontally to provide capacity for any c.f.m. requirement (see Fig. 191). The L-type has unit openings 20x20 inches and the rating is as follows:
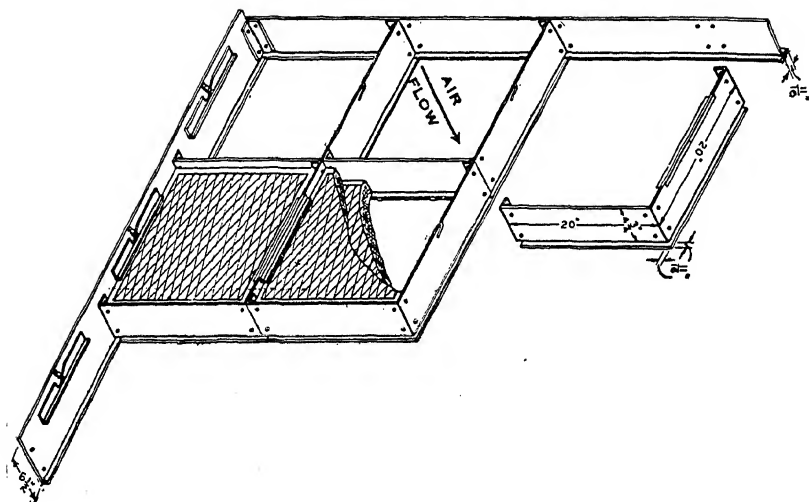
Fig. 190.   L-Type Filter Frame Assembly for Dust-Stop Filters
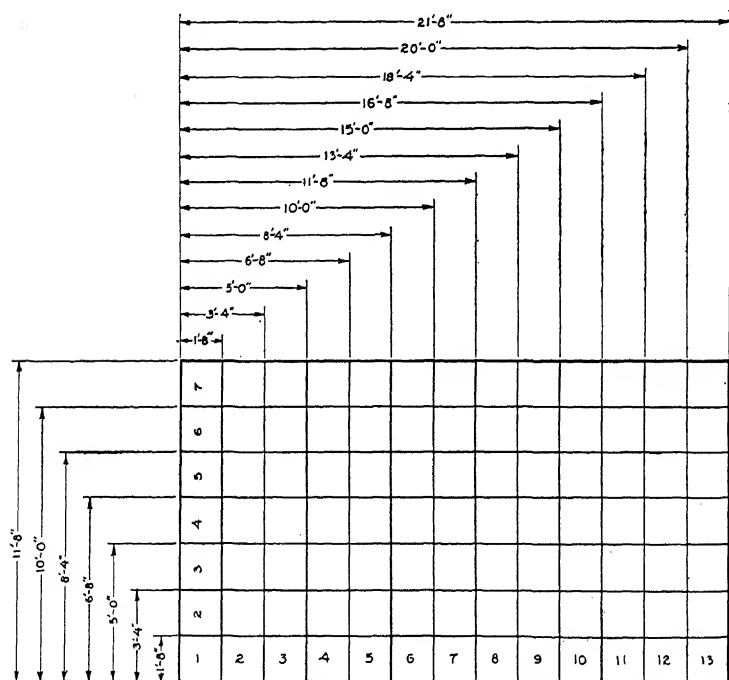
Fig. 191.   Dimensions Showing the Over-all Unit Measurements Not Including ⁵⁄₈-Inch Flanges on
Sides and Top

Capacity per unit (two filters in tandem)
800 c.f.m. at 300 f.p.m. velocity
Resistance at rated capacity:
Clean—.20 to .25 inches water
Loaded—.35 to .40 inches water
Efficiency—97% (A.S.H.V.E. method of rating).

V-Type Frame Assembly. Fig. 192 shows a V-type frame assembly. This frame has been designed for compactness and minimum of face area. Table 81 shows V-type data. (The dimensions do not include 1½-inch flanges on sides, top, and bottom.) The V-type frame is for filters 20x25 inches. The capacity for
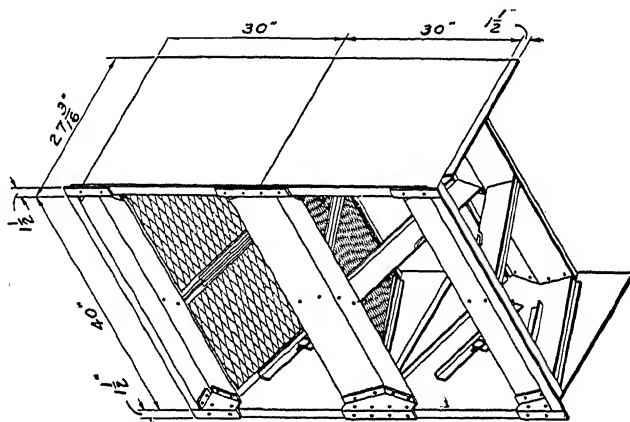
Fig. 192.   V-Type Filter Frame Assembly

each V of width and for each unit of height—2,000 c.f.m. at 300 f.p.m. velocity (two filters placed in tandem). The minimum combination, however, is a 1V-2 (one V wide, two filters high) having 4,000 c.f.m. capacity. See Table 81. Resistance at rated capacity is as follows:

Clean—   .22 to .25 inches water
Loaded—   .35 to .40 inches water
Efficiency—95% (A.S.H.V.E. method of rating)

The details of filter installation are identical in both the L- and V-type frames. Two Dust-Stop filters placed in tandem (front to back) are securely held in place against the gasketed frame by a progressive-wedge locking device. Both types are used for central heating, ventilation, air conditioning, and industrial processing applications.

In order to reduce resistance to an absolute minimum and to insure longest filter life, it is determined that 5 square inches of filter area should be provided for each 1 square inch of warm-air pipe area, or rated capacity of furnace in square inches. To determine the number of Dust-Stop filters required on a gravity warm-air furnace installation, first total the combined area in square inches of all warm-air pipes leading from the furnace, or rated capacity of furnace in square

inches, and multiply by five. This will give the *area* required in filters. Then, to determine the *number* of Dust-Stop filters required, divide by the area of the standard filter selected from Table 82.

Table 81. V-Type Data

Cubic Feet Per Minute	Combination	Number of Units or Coils	Width in Inches	Height in Inches	Depth in Inches	Cubic Feet Per Minute	Combination	Number of Units or Coils	Width in Inches	Height in Inches	Depth in Inches
4000	1 V-2	4	30	40	27 3/16	16000	2 V-4	16	60	80	27 3/16
6000	1 V-3	6	30	60	27 3/16	20000	4 V-2	16	120	40	27 3/16
8000	1 V-4	8	30	80	27 3/16		2 V-7	28	60	140	27 3/16
	2 V-2	8	60	40	27 3/16		7 V-2	28	210	40	27 3/16
10000	1 V-5	10	30	100	27 3/16	32000	4 V-4	32	120	80	27 3/16
12000	1 V-6	12	30	120	27 3/16	40000	4 V-5	40	120	100	27 3/16
	2 V-3	12	60	60	27 3/16		5 V-4	40	150	80	27 3/16
	3 V-2	12	90	40	27 3/16	50000	5 V-5	50	150	100	27 3/16
14000	1 V-7	14	30	140	27 3/16	56000	4 V-7	56	120	140	27 3/16
							7 V-4	56	210	80	27 3/16

Table 82. Areas and Circumferences of Warm= and Return=Air Pipe in Common Use—Showing Number of Dust=Stop Filters Required

Diam. Inches	Area Sq. In.	Circum. Inches	Number Required	Diam. Inches	Area Sq. In.	Circum. Inches	Number Required
7	38.48	21.99	1	22	380.1	69.12	4
8	50.26	25.13	1	24	452.4	75.40	5
9	63.62	28.27	1	26	530.9	81.68	6
10	78.54	31.42	1	28	615.8	87.97	7
12	113.1	37.70	2	30	706.9	94.25	9
14	153.9	43.98	2	32	804.3	100.5	10
16	201.1	50.27	3	34	907.9	106.8	11
18	254.5	56.55	3	36	1018.	113.1	12
20	314.2	62.83	4	..	......	......	..

Standard Dust-Stop Filters: 20x20x2"—400 sq. in.; 16x25x2"—400 sq. in.; 16x20x2"—320 sq. in.

Example. A gravity warm-air installation having four warm-air runs or leaders is as follows—

1 leader 8 inches..............area	50 square inches
1 leader 9 inches..............area	63 square inches
1 leader 10 inches..............area	78 square inches
1 leader 12 inches...............area	113 square inches
Total pipe area.............	304 square inches

Just for illustrative purposes the 16×25×2 inch filter (Table 82) will be used.

$$304 \times 5 = 1520 \text{ square inches}$$

The area of the 16×25×2 inch filter is 400 square inches. Then $1520 \div 400 = 3.8$ filters required. The fraction is considered as an additional filter so 4 filters would be required.

Self-Cleaning Filters. Fig. 193 shows a group of Double-Duty self-cleaning air filters. Fig. 194 shows a cross section of a Double-Duty Model L self-cleaning
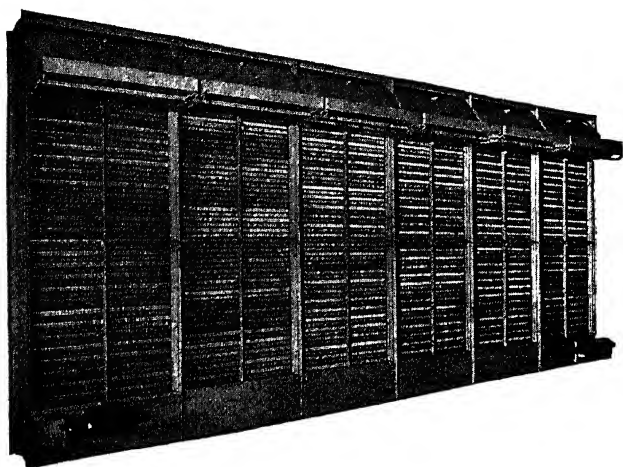
Fig. 193. Several Sections of Double-Duty Independent Self-
Cleaning Air Filter
Courtesy of Independent Air Filter Company, Chicago
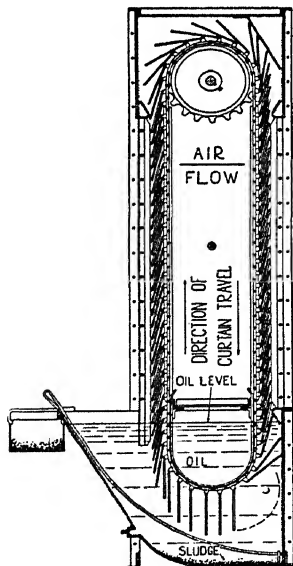
Fig. 194. Cross Section of a
Double-Duty Model L Self-
Cleaning Air Filter
*Courtesy of Independent Air
Filter Company, Chicago*

filter. This filter is primarily adapted to large-capacity, heavy-duty jobs. It is used for general ventilation in large buildings and for a wide variety of industrial applications—in fact, wherever conditions indicate the need for a strictly automatic, high efficiency filtering machine.

Like other Double-Duty filters, Model L operates on a principle of impingement. No filter using oil as the binding agent has ever proved successful unless adhering strictly to the basic principle of true impingement.

Double-Duty Model L is made in sizes ranging from single sections of 5,000 cubic feet per minute up to combination groups having a total capacity as high as 300,000 cubic feet per minute. The outstanding feature of the Double-Duty
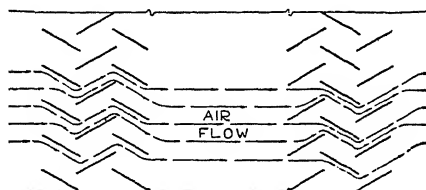
Fig. 195.　Diagram of Air Travel through Double-Duty Filter Curtain

filter is a pressed steel louver-plate curtain, which maintains its high initial efficiency at unvarying resistance under any and all conditions. It is entirely automatic in operation, requiring only occasional inspection and removal of sludge.

Two complete filtering actions take place through curtains which have an air space between them as shown in Fig. 195. The design of the individual baffle plates, with their rasp-like surfaces, gives the cleaning a high efficiency in this type of viscous impingement filter. The filter plates may be cleaned by long immersion in an oil bath to release the dust particles. The plates emerge cleaned and filmed with oil. Surplus oil always drains downward in such filters thus assuring against entrainment. The air, in passing through the filter, is given three acute deflections on each passage through the curtain—six deflections in all. See Fig. 195. Sludge should be removed at intervals of three to six months depending on dust content of the air.

Table 83 shows capacities and dimensions of Independent Double-Duty Filters.

To secure good air distribution select a group from those within the heavy lines. The groups shown outside the heavy lines are printed for reference only and are not in proportion to give best results. Always use the tallest filter that will fit in the space available.

Motors. Present-day air-conditioning apparatus is designed and driven almost entirely by electrical motors. This text does not go into the principles of motor design or operation as that is a complete subject in itself. However, the general method of selection of motors, which is the main interest of air-conditioning engineers, is explained briefly.

Table 83. Over=All Width of Independent Filters

Over-All Height	2'0"	3'0"	4'0"	5'0"	6'0"	7'0"	8'0"	9'0"	10'0"	11'0"	12'0"
5'9"	3000	4500	6000	7500	9000	10500	12000	13500	15000	16500	18000
6'3"	3500	5250	7000	8750	10500	12250	14000	15750	17500	19250	21000
6'9"	4000	6000	8000	10000	12000	14000	16000	18000	20000	22000	24000
7'3"	4500	6750	9000	11250	13500	15750	18000	20250	22500	24750	27000
7'9"	5000	7500	10000	12500	15000	17500	20000	22500	25000	27500	30000
8'3"	5500	8250	11000	13750	16500	19250	22000	24750	27500	30250	33000
8'9"	6000	9000	12000	15000	18000	21000	24000	27000	30000	33000	36000
9'3"	6500	9750	13000	16250	19500	22750	26000	29250	32500	35750	39000
9'9"	7000	10500	14000	17500	21000	24500	28000	31500	35000	38500	42000
10'3"	7500	11250	15000	18750	22500	26250	30000	33750	37500	41250	45000
10'9"	8000	12000	16000	20000	24000	28000	32000	36000	40000	44000	48000
11'3"	8500	12750	17000	21250	25500	29750	34000	38250	42500	46750	51000

Most units of air conditioning such as compressors, fans, washers, etc., have motors as an integral part of the units. The air conditioning engineer is not confronted with their selection except in very large systems. In such cases, motor manufacturers gladly take over the motor design or selection.
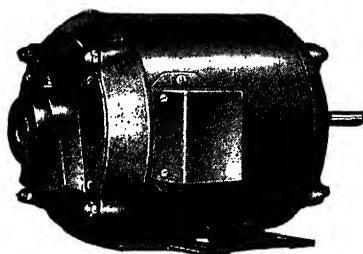
Fig. 196. Split-Phase Motor
Courtesy of Century Electric Co., St. Louis, Mo.

Fig. 196 shows a typical type motor called a split phase. These are adapted to drive domestic and industrial apparatus that do not require static or initial starting torque appreciably in excess of full-load torque—such as oil burners, unit heaters, air washers, blowers, fans, etc.

Note: Review what is given in chapter on "Fans" relative to horsepower, speed, etc.

Table 84. For Typical Motors for Compressors, Pumps, Fans, Blowers, Refrigerators, Stokers, Oil Burners

Type of Motor	Horsepower Range	Starting Duty	Remarks	Applications
SINGLE-PHASE MOTORS				
RS—Brush-Lifting	1/8 to 40	Heavy	Low Starting Current, High Starting Torque	Piston or Plunger Pumps, Refrigerators, Stokers, Compressors, etc.
BR—Brush-Riding	1/8 to 3/4	Heavy	Short Annual Service Characteristics	
CPH—Cap. Start and Run	1/8 to 10	Heavy	High Starting Torque	
CSH—Cap. Start	1/8 to 3/4	Heavy	High Starting Torque	
CSN—Cap. Start	1 to 10	Medium	High Starting Torque	Fans (Belted or Direct Connected) Centrifugal Pumps, etc.
CPX—Cap. Start and Run	1 to 10	Light	Must be Loaded to at Least 50% Capacity	
SP—Split Phase	1/60 to 1/3	Medium	Unrestricted Starting Current, Long or Short Annual Service Characteristics	Oil Burners, Unit Heaters, Blowers, Fans, Small Tools, etc.
SP—Split Phase	1/60 to 1/3	Light	Restricted Starting Current, Long or Short Annual Service Characteristics	
POLYPHASE MOTORS				
SC—Squirrel Cage	1/8 to 600	Medium	Normal Starting Current Normal Torque	General Purpose Motors
SCN—Squirrel Cage	7½ to 200	Medium	Lower Starting Current than SC Normal Torque	
SCH—Squirrel Cage	3 to 200	Heavy	Low Starting Current High Starting Torque	Refrigerators, Piston or Plunger Pumps, Compressors, etc.
AS—Automatic Start	1 to 60	Heavy	Lower Starting Current than SCH High Starting Torque	
SR—Slip Ring	1/2 to 200	Heavy	For Frequent Starting and/or Speed Control	Fans, Blowers, Centrifugal Pumps, Compressors, etc.

DIRECT-CURRENT MOTORS

Type of Motor	Horsepower Range	Remarks	Applications
DM-DN-R Shunt Wound Constant Speed	1/20 to 400	Torque is limited only by commutation. A direct-current motor has ample torque to start any load that it can carry when up to speed. Starting current is limited by controller to about 150% of full load current for light starting torque requirements with corresponding increases in current for increased starting torque	Fans, Blowers, Centrifugal Pumps, Machine Tools, etc.
DM-DN-R Compound Wound Varying Speed	1/12 to 400		Reciprocating Pumps, Compressors and Machines with Flywheels, etc.
DN-R Shunt Wound Adjustable Speed	1/2 to 100		Fans, Blowers, Machine Tools, etc.

Courtesy of Century Electric Co., St. Louis, Mo.

Table 84 gives recommendations for the selection of single-phase, polyphase, and direct-current motors in air-conditioning work.

*Refrigeration. Before refrigerating machines and other refrigeration processes can be thoroughly understood, it is necessary that the reader understand some of the basic principles.

There is a definite relationship between pressure and the boiling point of a liquid. The higher the pressure on a liquid, the higher its boiling point becomes. The boiling point of water is thought of as being 212°F., but this holds true only at sea level (29.92 inches of mercury). At the top of a mountain, water boils at much lower temperatures. Or again, water in a closed pan, under a high pressure of, say 100 pounds, boils at 338°F. Ammonia, sulphur dioxide, carbon dioxide, Freon, etc., boil at lower boiling points than water.

When water boils it evaporates. It takes up or absorbs heat from surrounding material. A liquid at a pressure of 58 pounds, for example, boils at 60°F., and if a pressure of 58 pounds is maintained within the pan or vessel that contains it, the liquid will boil if the surrounding air is higher than 60°F., and it will absorb heat from the air. Thus the boiling liquid absorbs heat from the air and cools it. This is what happens in the low pressure side of a refrigerating machine.

If some gas is put in a cylinder and compressed, its temperature rises. Thus if a common gas, such as Freon, is compressed from 47 pounds pressure and 60°F. to 120 pounds pressure without heat being added, its temperature will rise to 130°F. Then the gas can be condensed into a liquid at the higher pressure by a comparatively warm substance, such as water. In condensing a pound of this gas to a liquid, about 160 B.t.u.'s are removed from the gas. Compressing the gas, raises its heat content to a higher level, where it can be removed easily. The compressor (high side of a refrigerating machine) acts like a heat pump. A heat pump is a device that raises the temperature of a substance without addition of heat, so that most of the original heat can be removed by water. A condensing unit is such a device.

A refrigerant, in the form of a gas, is compressed with resultant rise in temperature. It then passes to a condenser where it can be condensed into a liquid at a high pressure by water. If this high pressure liquid is now allowed to expand through a throttling valve

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down to a lower pressure, some of it vaporizes at the expense of the heat contained in the liquid and produces a mixture of low temperature gas and low temperature liquid. If this liquid is allowed to pass to a coil (evaporator), it boils at a low temperature and takes heat from the surrounding air. The vapor resulting from the evaporation is then drawn back into the compressor cylinder and the cycle repeated. (See Fig. 197.)
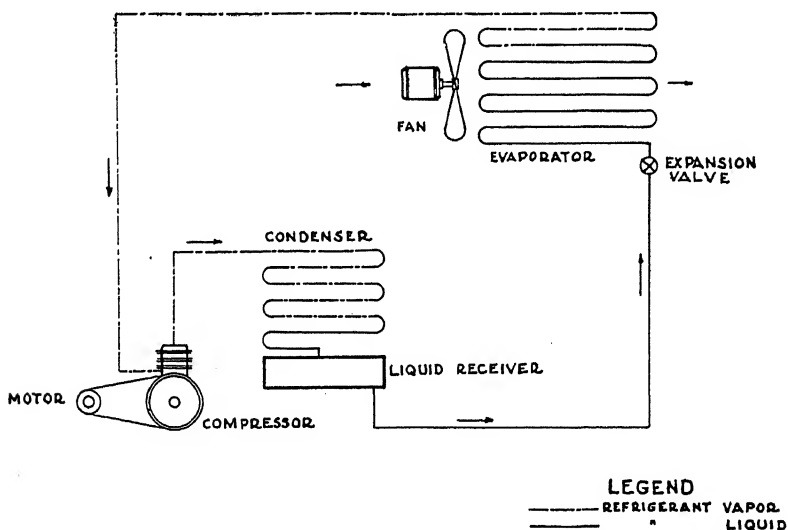
Fig. 197.   Diagram of a Condensing Unit and Evaporator as Used for Cooling Air
Courtesy of Bryant Heater Company, Cleveland, Ohio

The temperature of the refrigerant inside the evaporator determines the amount of heat that can be transferred to it. If a lower temperature is required to remove more heat, it is necessary to maintain a lower pressure in the evaporator. The lower this pressure, the lower the compressor efficiency.

Freon (F-12) is almost universally used as the refrigerant for this type of refrigeration machine, on account of its pressure-temperature characteristics and also on account of its non-toxicity. Freon is ideally suited for moderate-sized piston compressors, since the relation between latent heat and unit volume of vapor is such that the volume of vapor which the compressor must handle for a given refrigeration load is moderate. The general characteristics of several refrigerating media are shown in Table 85.

Table 85. *Properties of Common Refrigerants

	F-12	NH₃	SO₂	CH₃Cl	CO₂	CH₂Cl₂
Boiling Point at 1 atm. (°F)	−21.5	+28.0	14.0	−10.7	−108.4	10.5
Specific Gravity of vapor (Air=1)	4.16	0.596	2.264	1.784	1.529	3.00
Specific heat at constant pressure (C_p)	0.165	0.52	0.154	0.24	0.21	0.154
Specific heat at constant volume (C_v)	0.147	0.40	0.123	0.20	0.16	0.128
Pressure at 5°F.	11.9 lbs.	19.6 lbs.	5.9 In.	6.2 lbs.	319.7 lbs.	27.6 In.
Pressure at 35°F.	32.6 lbs.	51.6 lbs.	9.6 lbs.	24.3 lbs.	511.7 lbs.	9.5 In.
Pressure at 60°F.	57.7 lbs.	92.9 lbs.	26.2 lbs.	46.9 lbs.	729.5 lbs.	....
Latent heat /lb. at 5°F. (B.t.u.)	69.5	56.5	170.7	178.5	115.3	162.0
Volume of vapor at 5°F. (C.f. per lb.)	1.485	8.15	6.66	4.53	0.267	49.9

*Courtesy of the Bryant Heater Company, Cleveland, Ohio.

Note: Pressures are in pounds per square inch gauge or in inches of mercury vacuum.
    F-12 (Freon) Dichlorodifluoromethane   CH₃Cl Methyl Chloride
    NH₃ Ammonia   CO₂   Carbon Dioxide
    SO₂ Sulphur Dioxide   CH₂Cl₂ (Carrene) Dichloromethane

A consideration of the actions of gases under varying pressures reveals that the weight of a given volume of gas varies with the pressure exerted on that gas. Thus, a cubic foot of Freon vapor at 35° weighs only about 2/3 as much as a cubic foot at 60°, because the gas is more dense at the higher pressure corresponding to the higher temperature. Therefore, a compressor drawing gas from an evaporator having a pressure corresponding to a temperature of 35°, will in a given time and at a given speed, handle only 2/3 as great a weight of Freon as a compressor drawing gas from an evaporator operating at a pressure corresponding to a temperature of 60°. It follows that the unit in the first instance will have only 2/3 as great a heat "pumping" or heat absorbing capacity as the unit in the second instance. It is evident, therefore, that the capacity of a condensing unit is dependent on the suction pressure at which it is to operate. This fact is of especial economic significance where the adsorption method for dehumidification is used in combination with the refrigeration method for cooling.

Cooling and dehumidifying equipment employing refrigeration is often referred to as indirect expansion and direct expansion and may assume one of several forms. In the central plant type using indirect expansion, the actual cooling and dehumidification of the air may be accomplished by passing it through sprays of cold water. In large central plants this type is generally used because the spray

chamber which functions as a cooler and dehumidifier in summer can act as a humidifier in winter. A typical refrigeration central plant system with controls is shown diagrammatically in Fig. 198. The spray chamber (air washer) in Fig. 198 operates exactly as explained for air washers beginning on page 309.

In the direct expansion system, which is generally used in smaller installations, the evaporator containing the refrigerant is in direct
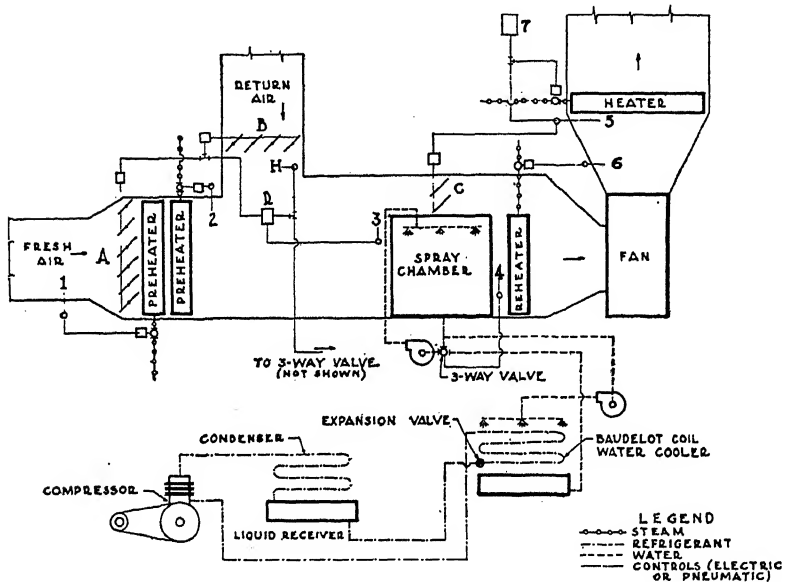
Fig. 198. Central Plant System—Refrigeration Only
Courtesy of Bryant Heater Company, Cleveland, Ohio

contact with the air circulated or is in the space from which the heat is to be removed. The evaporator is the heat transfer agent. The amount of heat absorbed will depend upon the average temperature of the refrigerant, the dry- and wet-bulb temperatures of the entering air, the velocity of air passing over the evaporator, and the thermal conductivity of the coil. This is explained under "Direct Expansion Coils."

The so-called "unit coolers," either floor or suspended type, with or without duct work, are classified as direct expansion systems. They consist essentially of an evaporator over which air is passed by a fan. If the dew point temperature of the air is **below** the

evaporator temperature, there will be no condensation of moisture and the heat removal will be entirely sensible heat. If the dew point of the air is **above** the evaporator temperature, there will be condensation of moisture and an increased amount of heat removal. Part of this will be latent heat and part sensible, the respective amounts being fixed by the characteristics of the evaporator. Further, there is a definite maximum proportion of the total heat removed by such an evaporator, which can be removed as latent heat.

Realizing the many advantages of independent control and recognizing that latent heat can be removed only at low suction pressures, and consequently at reduced compressor efficiency, some refrigeration engineers advise first removing sensible heat (accomplished at favorable suction pressures) and second, removing the latent heat separately. This scheme would call for multiple condensers and evaporators. But here again there is a definite maximum proportion of latent heat removal that can be secured, and if this maximum proportion is exceeded, reheating is required.

No such operating limitations exist where the adsorption method is used since that method can be designed to give any proportion of latent and sensible heat removal. A description of the adsorption method is given elsewhere in this section.

Refrigerating Machines. Where cold water is not obtainable, refrigeration equipment for the removal of sensible heat generally is recommended. This is usually of the electrically driven compression type, although absorption refrigeration, or steam ejector refrigeration, using steam from any source, may be used. It is also possible, when conditions permit, to use compression refrigeration with a gas engine as the motive power, or to drive the compressor by a synchronous motor where power factor correction is desired.

Fig. 199 shows one type of refrigerating compressor. Fig. 200 shows and explains the major operating parts. Table 86 gives typical ratings which, in this case, apply to the compressor shown in Fig. 199.

The type of compressor shown in Fig. 199 is used for unit coolers or air conditioners and in central systems where the requirements are not in excess of their highest ratings. They can be used for residential warm air-conditioning apparatus and for split systems. In fact, where refrigeration is needed, there is a place for them. The compressor shown in Fig. 199 is air cooled.
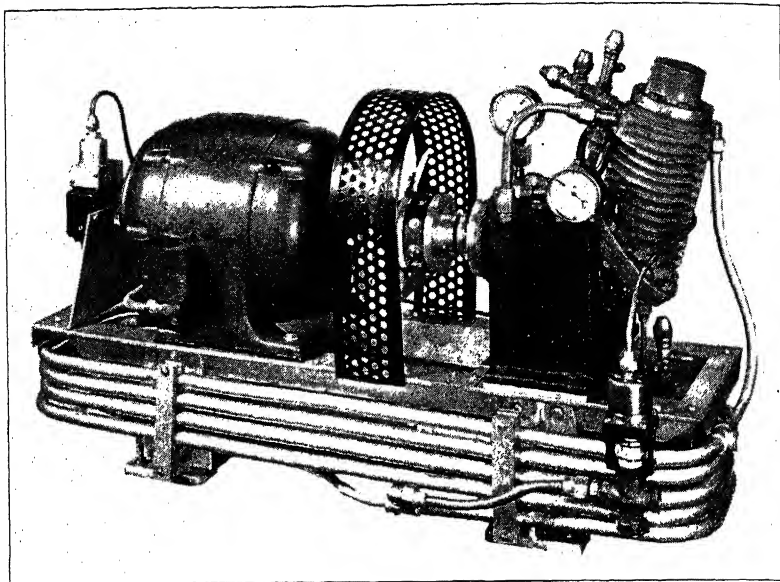
Fig. 199. Thermal Unit V-8 Refrigerating Compressor
Courtesy of Thermal Units Mfg. Co., Chicago

Table 86. Thermal Unit V=8 Compressor
Refrigerating Capacities with Freon Refrigerant—B.t.u. per Hour

Cond. Pressure Lb./Sq.In. Gauge	Cond. Temp. °F.	Suction Pressure—Lb. per Sq. In.—Gauge								
		0	5	10	15	20	25	30	35	40
		Suction Temperature—°F.								
		−21.7	−8.9	1.6	10.6	18.4	25.5	31.9	37.9	43.3
Speed: 1725 R.p.m. Motor: 1½ Hp. Model: 158										
60	61.9	5220	6950	8940	11010	13110	15100	17090	19110	21150
70	69.9	4980	6750	8570	10510	12590	14470	16330	18300	20150
80	77.2	4750	6460	8200	10090	12030	13870	15780	17690	19500
100	90.2	4350	5940	7550	9330	11130	12800	14450	16180	17900
120	101.7	3960	5450	7000	8600	10300	11910	13510	15130	16740
140	112.0	3530	4950	6540	8040	9560	11130	12660	14250	15800
Speed: 1150 R.p.m. Motor: 1 Hp. Model: 108										
60	61.9	3480	4640	5960	7350	8750	10030	11390	12730	14100
70	69.9	3320	4500	5720	7010	8390	9650	10900	12200	13420
80	77.2	3165	4310	5470	6725	8050	9250	10510	11800	13000
90	84.0	3030	4125	5245	6450	7700	8860	10000	11220	12400
100	90.2	2900	3960	5030	6220	7420	8530	9640	10790	11930
140	112.0	2355	3300	4360	5350	6375	7420	8440	9500	10530
Speed: 870 R.p.m. Motor: ¾ Hp. Model: 78										
60	61.9	2610	3500	4500	5500	6550	7550	8550	9560	10580
70	69.9	2500	3380	4285	5260	6290	7240	8170	9150	10075
80	77.2	2380	3230	4100	5045	6020	6985	7890	8840	9750
90	84.0	2280	3090	3930	4840	5780	6650	7500	8420	9300
100	90.2	2175	2970	3780	4670	5560	6400	7225	8090	8950

Fig. 201 shows a Carrier compressor of the type necessary where refrigeration in terms of tons is necessary for larger installations. More than one unit may be interconnected on a common suction for use on the same refrigeration load. This type of compressor is generally used in central systems, as explained under "Direct Expansion
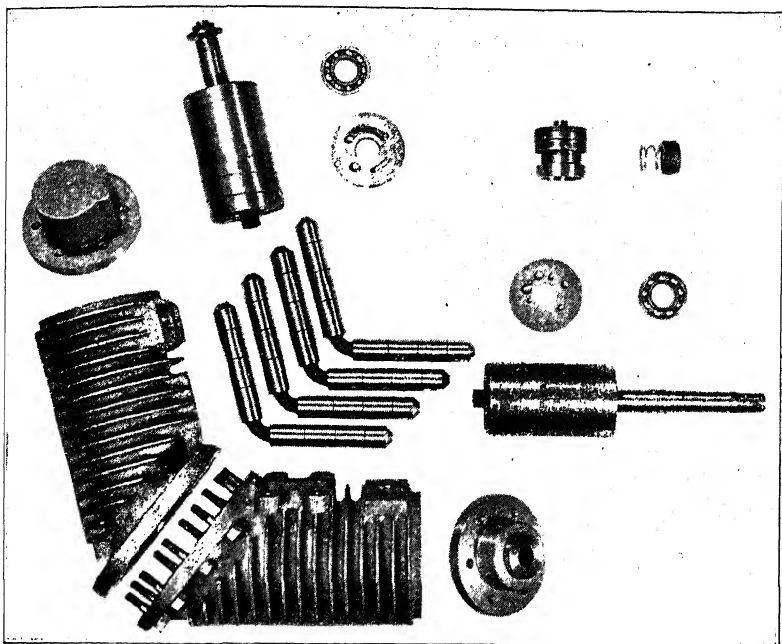
Fig. 200.   Major Operating Parts of Compressor Shown in Fig. 199
Courtesy of Thermal Units Mfg. Co., Chicago

Coils," and for systems of large characteristics where air washers are used. The large machines, as in Fig. 201, require water or evaporative condensers to remove the heat of condensation from the refrigerant gas.

Any recognized manufacturer of refrigerating equipment can specify the proper unit if he knows the amount of heat that is to be removed in the main cooler and the temperature of cooling water (if necessary) available for the condenser. He should also know the probable distance between compressor and evaporator, and that the unit is to operate in most cases with an evaporator temperature of not less than 55°F. or 60°F.

The curves in Fig. 202 show the capacities of typical Freon compressors at various evaporator or cooler temperatures when using 76°F. cooling water. For higher cooling water temperatures, the capacities are decreased slightly, and for lower cooling water temperatures they are increased slightly. Note that as the evaporator or cooler temperature is increased, the capacity of the compressors is increased. This is because the higher evaporator temperatures cor-
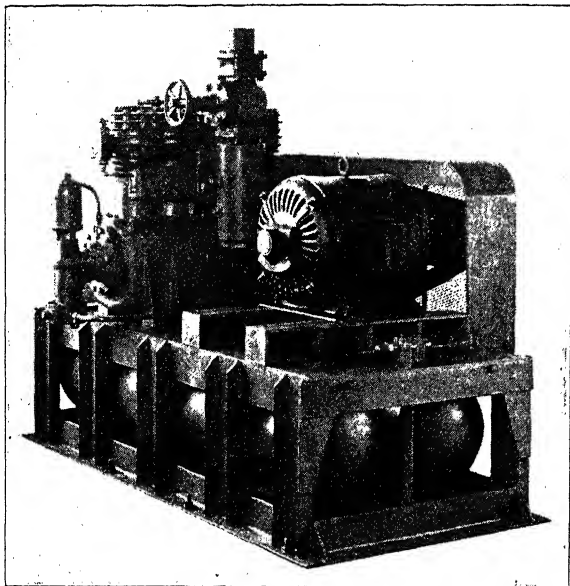
Fig. 201. Carrier Type 7G8 Refrigerating Machine
Carrier Engineering Corp., Newark, N. J.

respond to higher evaporator pressures which, in turn, yield a denser refrigerant vapor to the inlet of the compressors and allow them to pump a greater weight of refrigerant at each stroke. The condenser supplied with the compressor must have sufficient surface to remove the additional heat which is delivered by the increased weight of refrigerant.

Fig. 203 shows a diagrammatic arrangement of a centrifugal refrigerating machine used in providing large refrigerating capacities for commercial air-conditioning work.

The machines are available in various sizes, ranging in capacity from 50 to 850 tons, air-conditioning rating. The refrigerant is

Carrene No. 2 (trichloromonofluoromethane), non-toxic, non-explosive, non-inflammable, and highly efficient. In all sizes, the machines are compact self-contained units, comprising an evaporator centrifugal multi-stage compressor, and condenser on a single foundation. They are used to furnish chilled brine (salt or other solution or fresh water) to the process equipment or the air-conditioning apparatus, for direct cooling of beverages or condensation of gases. Machines
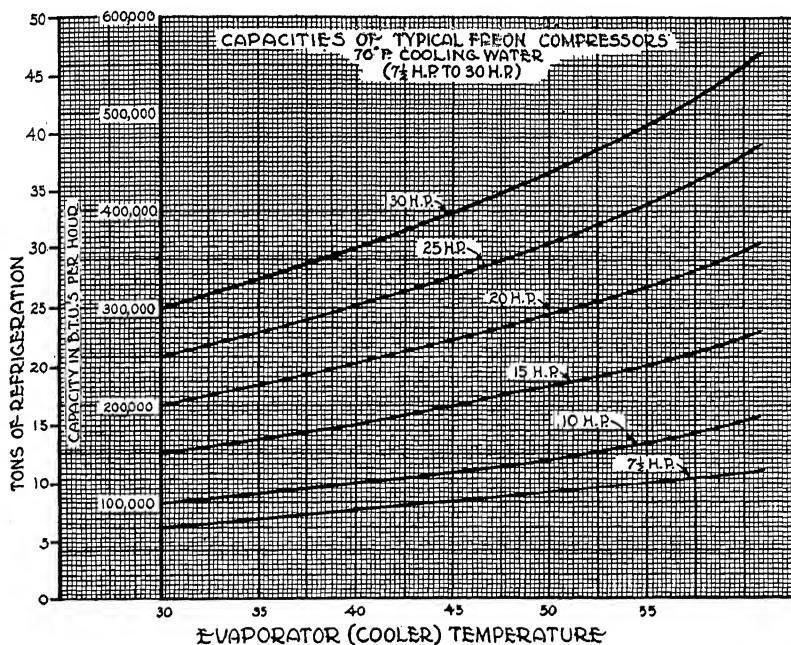
Fig. 202.   Capacities of Typical Freon Compressors

are built in various stages or stage combinations as required for special applications. For process work, temperatures as low as 100°F. are attained with these machines, but for air-conditioning work the spray water usually is not taken below 40° F.

The machines may be located nearby or remotely from the point of heat absorption, consequently one machine or group of machines may serve the entire requirements for a large plant or building or industrial load.

The compressors are multi-stage centrifugal. The rotor is statically and dynamically balanced and the impellers are lead coated to

preserve this balance. The complete oiling system with pump is an integral part of the compressor, with the oil cooler located externally.

These compressors permit wide selectivity in the following types of drive: (1) synchronous motor, constant speed, used for power factor corrections; (2) induction motor, constant speed, of low initial cost but not permitting greatest economy of operation; (3) slipring motor, variable speed, manual or automatic control; (4) Diesel or gas-engine, variable or constant speed, particularly practical where electric rates are high but low cost gas is available; (5) steam turbine,

NOTE: BRINE AND WATER THROUGH TUBES,
LIQUID AND VAPOR OUTSIDE OF TUBES
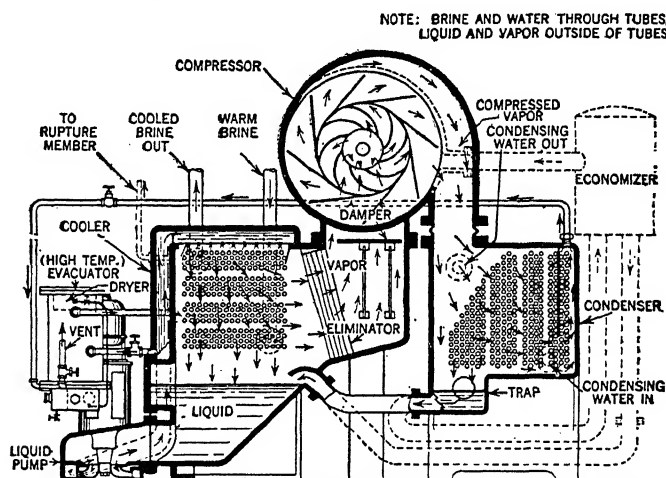
Fig. 203.   Diagrammatic Arrangement of a Carrier Centrifugal Refrigerating Machine

variable speed, manually or automatically controlled. This drive is most desirable as it combines low cost with a maximum of economy and flexibility. It may be driven by high pressure steam and supply low pressure steam for heating or other industrial purposes. Or it may utilize low pressure exhaust steam, operating condensing, or a combination of the two. The steam turbine exhausts clean steam, free from contaminating oil.

There are few moving parts and they are rotating, free from vibration and shock, light in weight, simple in construction; all bearings and parts are easily accessible; pump-down preceding maintenance work rarely is required; brass tubes less subject to corrosion pitting and fouling than steel tubes; oil does not come in contact with the refrigerant, so the cleaning of heat transfer surfaces is simplified.

The machine is safe and the refrigerant is safe. All moving parts are enclosed, and for usual temperature conditions both condenser and cooler operate either under a vacuum or slightly above atmospheric pressure.

Cycle of Operation. The warm brine to be cooled (salt solution or water) is circulated through the cooling coils, over which flows the liquid refrigerant, brought over the coils by the liquid pump. The liquid refrigerant is evaporated from the coils, taking up the heat of the brine, and drawn into the compressor. The compressed gases
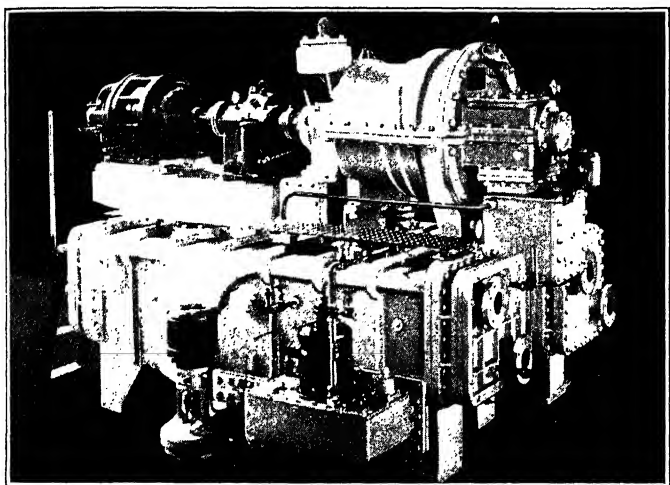
Fig. 204. Battery of Centrifugals for Office Building
Carrier Engineering Corporation, Newark, N. J.

are forced into the condenser, where the heat is removed by the condenser water coming from the city supply, deep well, or cooling tower flowing through the coils. The refrigerant becomes liquid again and flows back to the evaporative storage tank, completing the cycle. Fig. 204 shows a typical installation of centrifugal machines for an office building.

Vac=Coolers. Fig. 205 shows a *Vac-Cooler used in the air conditioning of large buildings, ballrooms, etc. Its principle of operation is the cooling of water by evaporation. This type of cooler is extremely economical to operate and has no moving parts except a few centrifugal pumps to handle the various water circuits. One of the impor-

*Courtesy of Westinghouse Electric & Mfg. Co. Mansfield, Ohio.

tant features of this type of cooler is that the larger sizes are equipped with multiple booster ejectors. This makes it possible to shut down a portion of them at partial load to effect savings, because economies at partial loads are practically equivalent to the economies at full load.

Booster Ejector. The booster ejector is probably the most important part of a Vac-Cooler. It makes possible starting without flash-back, stable operation, and high efficiency. The achievement of
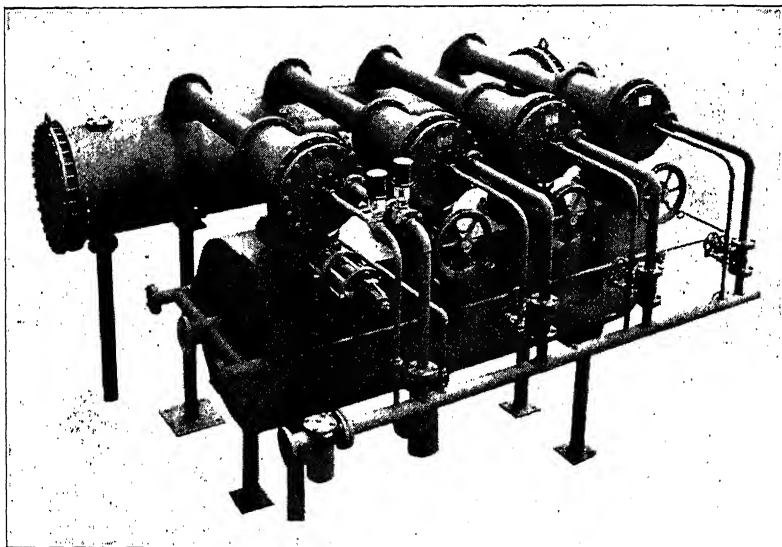
Fig. 205. Vac-Cooler
Courtesy of Westinghouse Co., Mansfield, Ohio

these necessary characteristics requires a unit, as shown in Fig. 206. The nozzles are mounted in nozzle holders and project past the vapor inlet to allow parallel flow of the vapors and high velocity steam jets. This design has been found to require the least steam per unit capacity when in operation, but it requires a momentary increase in steam flow when starting. The increase can be supplied momentarily and then shut off because of a single nozzle, extension, and steam chest. This is equivalent to the use of the choke on an automobile engine.

Suction Valves. There are three ways in which each booster ejector may be segregated to operate the unit at partial load: (1) by compartments in the cold tank; (2) by compartments in the con-

denser; and (3) by suction valves at the inlet to each booster ejector. Of these, the use of suction valves will cause the least complication and provide for the greatest potential savings in steam and/or condenser water when the condenser water temperature or the load is less than the maximum for which the unit is designed.

On most air-conditioning and many industrial cooling jobs, it is desirable that the full quantity of chilled water be circulated continuously, regardless of load variations. Any attempt to divide the cold tank into compartments seriously interferes with and usually prevents this full circulation.
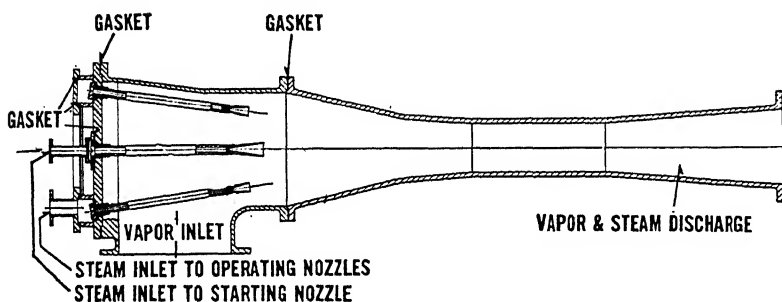
Fig. 206. Sectional View of Multiple Nozzle Booster Ejector

Compartments in the condenser prevent the use of the entire condenser surface when running on partial loads or with colder than maximum condenser water. Under either or both of these conditions, the condenser will give a higher vacuum than that for which the booster ejector exhaust condition is set. When this occurs, a considerable saving can be had, either by throttling steam to the boosters or by throttling water to the condenser, whichever produces the greatest saving. The use of suction valves leaves both the cold tank and the condenser free and open to accomplish the desirable functions.

Cycle of Operation. A schematic diagram of operation of a Westinghouse Vac-Cooler is shown in Fig. 207. Steam under pressure is delivered to the operation nozzles of the booster ejector. This steam is expanded through the nozzle and discharged into the booster ejector at high velocity. This high velocity steam jet entrains water vapor from the cold tank and compresses it to such a pressure that the mixture can be condensed in the condenser.

The water to be chilled is continuously sprayed into the cold
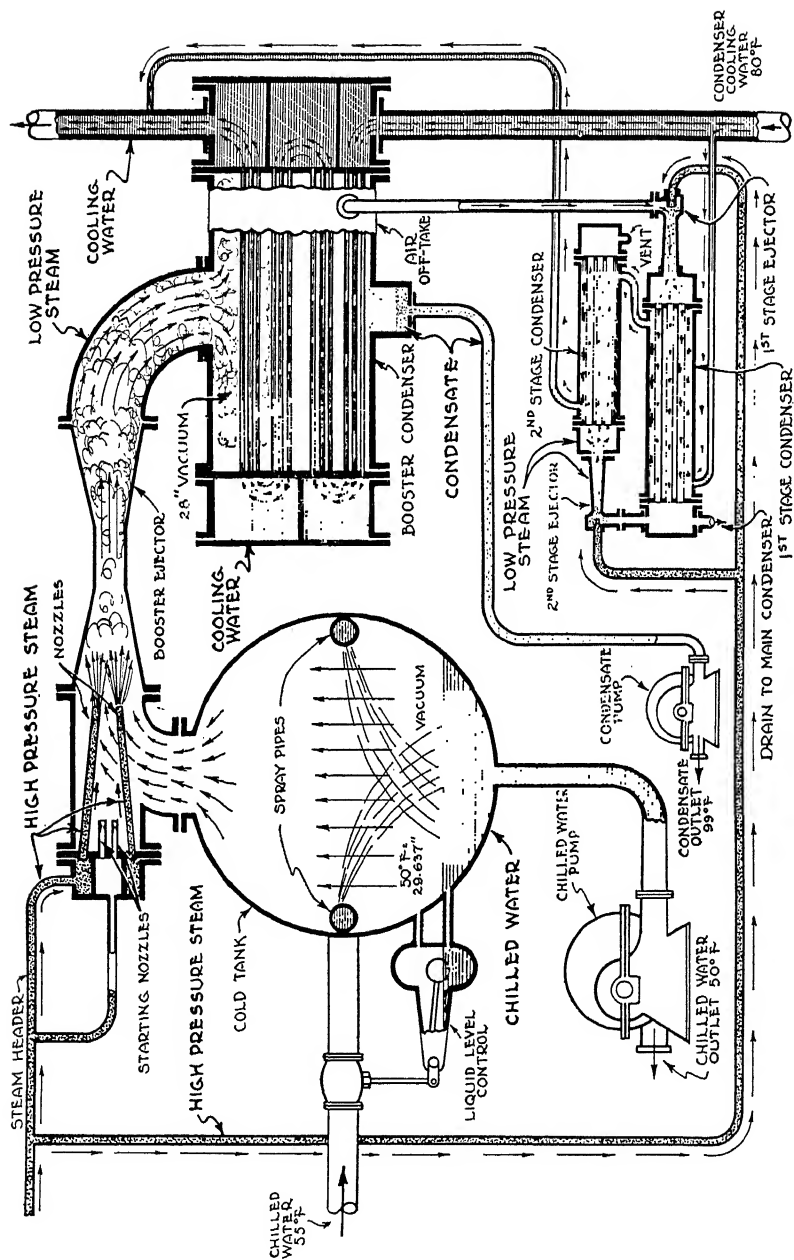
tank in thin sheets throughout the length of the tank. A liquid level controller regulates the quantity entering the cold tank and maintains a substantially constant level in the cold tank. A chilled water pump circulates the cooler water through the external cooling equipment and back into the cold tank.

A multi-pass type condenser serves the booster ejectors and is in turn served by an air removal apparatus which usually consists of a first and second stage steam jet air ejector with intermediate and after condenser.

Note that steam and cooling water pass counterflow through these condensers. Also note that the air offtake from the main condenser is situated at the coldest part of the condenser, thus reducing to a minimum the capacity requirements of the two-stage air ejector apparatus. The first stage condenser drains its condensate into the main condenser through a loop seal. A condensate pump removes the condensed vapors of the booster ejector and discharges them into the sewer or into boiler feed.

Ratings. Vac-Coolers are made in the typical sizes shown in Table 87. The units specified as special non-flexible units are somewhat lower in first cost than the flexible type. The units are available for steam pressures from 1 pound to 250 pounds gauge and for condenser water temperatures from the lowest temperatures available for such service up to 95°F. in standard ratings and sizes. The standard ratings of Vac-Coolers are based on 50°F. chilled water temperature leaving the unit. The capacities of these units with other chilled water temperatures are given in Table 88. A study of the characteristics of the Vac-Cooler in Table 88 for, let us say, 14 tons capacity at 35°F., shows that at 45°F. the capacity is 21 tons, at 55°F. is 29 tons and for 70°F. is 44 tons. The logarithmic average capacity over this temperature range is 26.2 tons, or approximately double the capacity at 35°F. A great reduction may be made in first cost and operating cost by recycling water through the unit and the storage tank until the temperature is reduced to 35°F.

Ton of Refrigeration. The capacities of most refrigerating machines are expressed in terms of tons of refrigeration. Also when calculating cooling requirements the ultimate requirement is often given in terms of tons of refrigeration. In expressing refrigerating capacities or requirements each ton is thought of as equal to the

Table 87. Typical Sizes of Vac–Coolers

Tons Cap.	Std. No. Boosters	Spec. Non-Flexible Units	Tons Cap.	Std. No. Boosters	Spec. Non-Flexible Units	Tons Cap.	Std. No. Boosters	Spec. Non-Flexible Units	Tons Cap.	Std. No. Boosters	Spec. Non-Flexible Units
15	1		100	3	2	250	4	.....	600	6	.....
25	2	1	125	3	.....	300	4	.....	700	6	.....
40	2	1	150	3	.....	350	4	.....	800	6	.....
50	2	1	175	3	.....	400	4	.....	1000	6	.....
75	2	1	200	4	.....	500	5	.....	1250	6	.....

Table 88. Westinghouse Standard Vac–Coolers

Chilled Water Temp.	Frame Factor	15 Tons	25 Tons	40 Tons	50 Tons	75 Tons	100 Tons	125 Tons	150 Tons	175 Tons	200 Tons	250 Tons	300 Tons	350 Tons
35	.55	8	14	22	28	41	55	69	83	96	110	138	165	193
36	.58	9	15	23	29	44	58	73	87	101	116	145	174	203
37	.61	9	15	24	31	46	61	76	92	107	122	153	183	214
38	.64	10	16	26	32	48	64	80	96	112	128	160	192	224
39	.67	10	17	27	34	50	67	84	101	117	134	168	201	235
40	.70	11	18	28	35	53	70	88	105	123	140	175	210	245
41	.73	11	18	29	37	55	73	91	110	128	146	183	219	256
42	.76	11	19	30	38	57	76	95	114	133	152	190	228	266
43	.79	12	20	32	40	59	79	99	119	138	158	198	237	276
44	.82	12	21	33	41	62	82	103	123	144	164	205	246	287
45	.85	13	21	34	43	64	85	106	128	149	170	213	255	298
46	.88	13	22	35	44	66	88	110	132	154	176	220	264	308
47	.91	14	23	36	46	68	91	114	137	159	182	228	273	319
48	.94	14	24	38	47	71	94	118	141	165	188	235	282	329
49	.97	15	24	39	49	73	97	121	146	170	194	242	291	340
50	1.00	15	25	40	50	75	100	125	150	175	200	250	300	350
51	1.03	15	26	41	52	77	103	129	155	180	206	258	309	360
52	1.06	16	27	42	53	80	106	132	159	186	212	265	318	371
53	1.09	16	27	44	55	82	109	136	164	191	218	273	327	382
54	1.12	17	28	45	56	84	112	140	168	196	224	280	336	392
55	1.15	17	29	46	58	86	115	144	173	201	230	288	345	403
56	1.18	18	30	47	59	89	118	148	177	207	236	295	354	413
57	1.21	18	30	48	61	91	121	151	182	212	242	303	363	424
58	1.24	19	31	50	62	93	124	155	186	217	248	310	372	434
59	1.27	19	32	51	64	95	127	159	191	222	254	318	381	445
60	1.30	20	33	52	65	98	130	163	195	228	260	325	390	455
61	1.33	20	33	53	67	100	133	166	200	233	266	332	399	465
62	1.37	21	34	55	69	103	137	171	206	240	274	343	411	480
63	1.41	21	35	56	71	106	141	176	212	247	282	353	423	494
64	1.45	22	36	58	73	109	145	181	218	254	290	363	435	508
65	1.50	23	38	60	75	113	150	188	225	263	300	375	450	525

absorption of the heat given up by one ton of ice at 32°F. melting to water at 32°F. in 24 hours. This is the same as heat absorption at a rate of approximately 200 B.t.u.'s per minute or 12,000 B.t.u.'s per hour. (See also Chapter XVII.)

Selection of Refrigerating Machines. The selection of a refrigerating machine can be made according to tons of refrigeration needed or B.t.u.'s to be removed. Most manufacturers' catalogues rate various

machines in terms of tons of refrigeration. Others rate the machines according to B.t.u.'s per hour. (Table 86.) Two factors control the size of the refrigeration system, (1) the evaporator or suction temperature and (2) the condenser or head temperature. (See Table 86.) With the knowledge that the system will operate most of the time with a load of not over 60 or 65% of maximum, and that maximum demands will occur at infrequent times and for only short periods, some provision must be made to insure economical operation under average conditions. This can be done by overloading the machine under highest demands and basing the design on average loads. It is comparatively easy to furnish condensers and evaporators to carry the maximum load so arranged that they will function properly at small demands. They affect the compressor performance to some extent but most of the compressor problems are in the machine itself.

Variations in load are usually effected by lowering the suction temperature and pumping a larger volume of gas per ton through a greater pressure range. This is possible because the latent heat of the refrigerant remains nearly constant throughout the small range used and the specific volume varies rapidly with change in pressure. As the compressor must remove the refrigerant evaporated, the evaporator temperature fixes the displacement required. The objection to such method is that the total power consumed remains nearly constant and the power per unit of cooling increases rapidly as the total output is reduced. Such operation is satisfactory as long as the load is kept within 10% of the rating of the compressor but this condition does not commonly occur in air-conditioning applications.

*Silica Gel Dehumidifier. Silica gel is a hard, hygroscopic, manufactured substance, similar in appearance to clear quartz granules, but vastly different structurally. Silica gel granules are extremely porous. Silica gel can be visualized as a body of ultra-microscopic particles piled in a homogeneous mass similar to a pile of cannon balls or a shredded wheat biscuit, but the voids in silica gel are about a billionth the size of the voids in these two examples. Thus an infinitesimal mass of pores can be imagined. Silica gel employs the principle of capillary attraction—the power of liquids to rise in tubes —the principle is demonstrated by the flow of ink up the tubes in a

*Data by Courtesy of the Bryant Heater Co., Cleveland, Ohio.

blotter. It is evident that if an extremely small capillary tube is brought into an atmosphere of condensable vapors, in which the vapor pressure is higher than that which would be in equilibrium with the liquid contained in the capillary, then condensation of the vapor in the capillary tube will take place.

These pores have intense capillary action and when moist air is brought into contact with the substance the pores absorb not only water but also vapors and hold both as a sponge holds water. (Yet the silica gel to all appearances is perfectly dry.) Normal commercial silica gel will absorb approximately 40% of its weight of water from saturated air. When the silica gel becomes saturated and can absorb no more water, it can be reactivated by the application of heat which dries the substance by evaporation. This operation can be repeated indefinitely without affecting its adsorption efficiency. A point to be emphasized here is that silica gel has an actual affinity for water and that whenever water contacts the substances adsorption takes place by natural law.

Another point to be remembered is that silica gel produces or causes adsorption entirely independent of temperature. In this way it is different from an air washer where very low temperatures are sometimes required to accomplish the desired dehumidification.

When moisture is adsorbed by silica gel, heat is released to the air passing through it equal to the amount which would be required to evaporate an equal amount of water. Thus dehumidification by adsorption produces heat. In air-conditioning systems this heat must be removed by coolers, just as humidification requires heat which must be added to the load of a heating plant. This heat, however, is at a relatively high level so that it can be removed readily and economically by cooling media. In most cases ordinary tap water can be used for this purpose. When refrigeration is used for cooling purposes, in conjunction with silica gel dehumidification, the expansion coil usually need not be kept at a lower temperature than 60°F.

Referring to the group of constant pressure curves shown in Fig. 208, it can be noted that the adsorptive capacity of silica gel increases with a lower temperature, or with an increased vapor pressure or moisture content in the air.

For example, an inspection of these curves shows that when silica gel is in equilibrium at a temperature of 80°F. with water vapor

at 59° dew point or at a vapor pressure of .5 inches mercury, the corresponding pressure is $\frac{5}{1}$ (the vapor pressure of water at 80°F. being 1 inch mercury) and the concentration in the silica gel is 31%.

The drying of air (dehumidification) by means of silica gel must, therefore, be considered a physical phenomenon and not a chemical reaction. In other words, it does not change in appearance or struc-
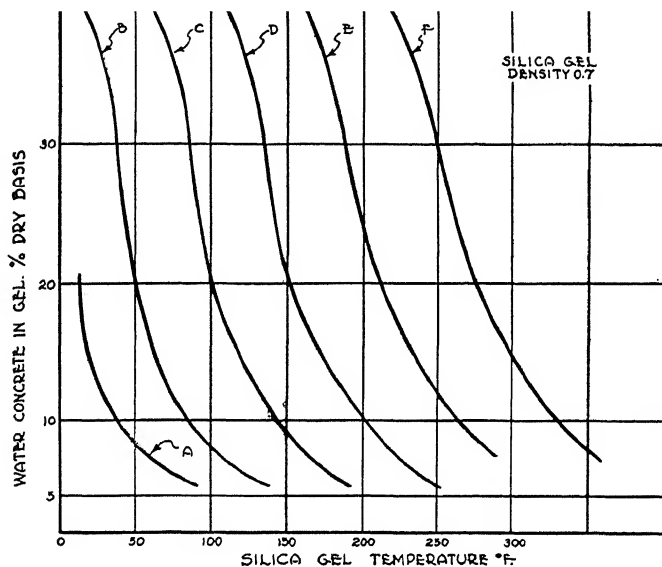
Fig. 208.   Equilibrium Curves of Silica Gel and Water Vapor

A = 0.001" HG. (−) 60°F. D.P.
B = 0.01" HG. 20°F. D.P.
C = 0.5" HG. 59°F. D.P.
D = 2" HG. 101°F. D.P.
E = 8" HG. 152°F. D.P.
F = 30" HG. Atmos. 212°F. D.P.

Courtesy of Bryant Heater Company, Cleveland, Ohio

ture as it adsorbs water vapor. The removal of water vapor is practically instantaneous.

The saturation or capacity of silica gel to hold water varies with the vapor pressure of the moisture in the air.

The amount of heat required to activate silica gel varies with the design of the equipment. This is because additional heat is required to raise the temperature of the metal supporting the substance and to compensate for heat loss due to radiation. In the equipment explained later in this chapter the heat for activation is obtained by

gas burners. The products of combustion are mixed directly with the activating air. Oil or a high pressure steam coil may also be used.

The equipment required to dehumidify air by adsorption consists of silica gel supported in suitable beds, fans to convey the adsorption and reactivating air through the beds, a heater, motor, cooler, and suitable automatic controls. The completely detailed
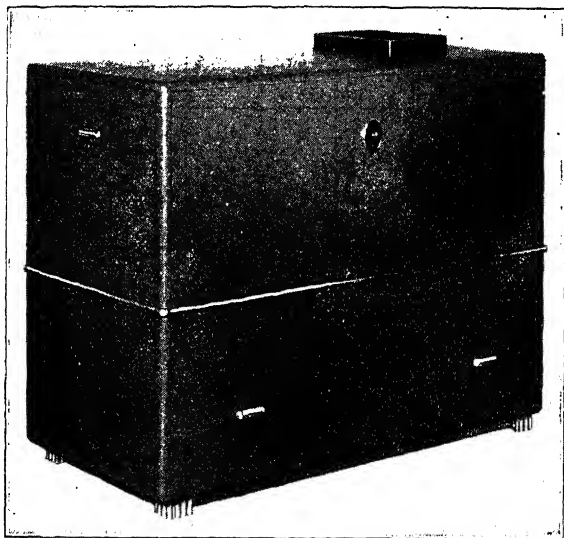
Fig. 209.   Silica Gel Apparatus
Courtesy of Bryant Heater Company, Cleveland, Ohio

description for silica gel dehumidifiers is given throughout pages 369 to 373 following.

Bryant Dehumidifier. Fig. 209 shows a Bryant dehumidifier as it appears completely assembled. Fig. 210 shows a phantom view of the same apparatus shown in Fig. 209. The following description explains all major parts and their operation.

Operation. A dehumidifier of the type shown in Fig. 209 is adapted primarily to central plants (where all conditioning is done for whole building at one point and then distributed by ducts). They deliver a constant supply of dry air when in operation and can be controlled automatically. The apparatus consists of two separate

compartments of silica gel adsorbing material. The air being treated is passed alternately through the two compartments. At the time one compartment is treating air, the other is being activated. The control is by dampers.

The two compartments A and B (Fig. 210) are located along the length of the machine. Each compartment consists of three or more separate beds in which the silica gel is supported between screens.
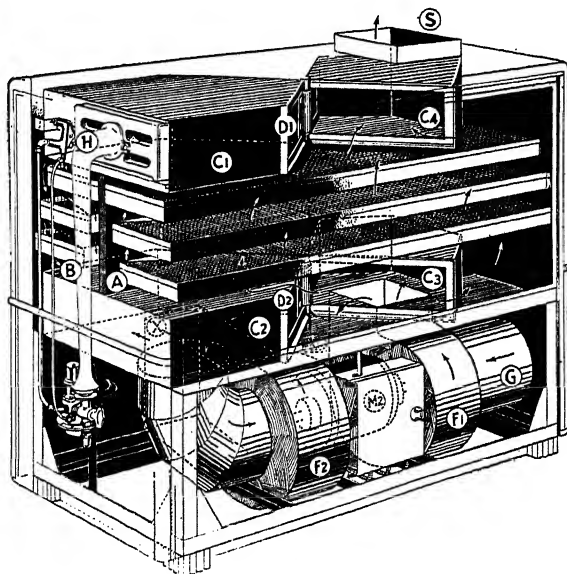
Fig. 210. Phantom View of Silica Gel Apparatus
Courtesy of Bryant Heater Company, Cleveland, Ohio

These beds are so arranged that part of the air passes through each one.

The air passes through the beds in an upward direction, when being dehumidified, and downward during activation. The counter-flow causes self-cleaning of the beds.

Contained within the compartments at both top and bottom, are two chambers in which are located twin two-position four-way dampers, D_1 and D_2 (Fig. 210) that control the passage of air through the two compartments. These chambers are designed as follows: C_1 activation inlet chamber; C_2 activation outlet chamber; C_3 adsorption inlet chamber; and C_4 dry (dehumidified) air outlet chamber.

The dampers, both in the same vertical plane and connected to a common shaft, open chambers C_1 and C_2 to either compartment A or B. Simultaneously chambers C_3 and C_4 are opened to either B or A.

Activation Cycle. Hot activation air from the gas burner H, Fig. 210, is drawn through chamber C_1 and is directed by damper D_1 downward through the silica gel compartment, past damper D_2 and into the activation outlet chamber C_2. From there it is drawn through fan F_2 to the exhaust stack, carrying with it the moisture which has been removed by the silica gel beds during the previous period.

Adsorption Cycle. The air to be dehumidified is drawn in through a duct from the outside to the adsorption inlet chamber C_3 past damper D_2, Fig. 210. It is forced upward through the silica gel compartment and guided by damper D_1 into the dry air outlet chamber C_4 from whence it goes to the supply duct going to the system.

Cycle Control. The dampers are shifted automatically once every ten minutes under the control of an electric timer. While one silica gel compartment is adsorbing for a ten-minute period, the other is being activated for seven minutes and "purged" for three minutes. The purging period is necessary to cool the silica gel to the point where it will again absorb water vapor. Purging is accomplished automatically by the electric timer, which turns off the gas burners and allows unheated air to pass through the silica gel beds.

The dampers are shifted by a rod and crank-arm linkage driven by a small geared head motor. The complete shift requiring from three to five seconds is started by a contact on the electric timer and is stopped by a limit switch actuated by the crank arm which breaks the electric circuit when the dampers have moved into their new position.

Fig. 211 shows a complete typical basement layout, in plan and elevation, of an installation which is year 'round in operation and which employs silica gel to dehumidify.

Summer Operation. In the summer, wet outdoor air is drawn through duct *1*, Fig. 211, dried in the dehumidifier, and delivered as dried warm air through duct *2* to the dry air cooler; from which it issues as dried cool air. This dried air is delivered to return duct *3*, which conveys the dried cool air to chamber *5* where it is mixed with the recirculated air from the premises being conditioned by return

duct *4*. The mixture is then filtered and passed through the main cooler. A fan located in chamber *6* delivers the cool, dehumidified, and filtered air to furnace casing *7*, whence it is conducted by the supply duct *8* to the premises being conditioned.

Connection *A* permits some recirculated air being drawn from
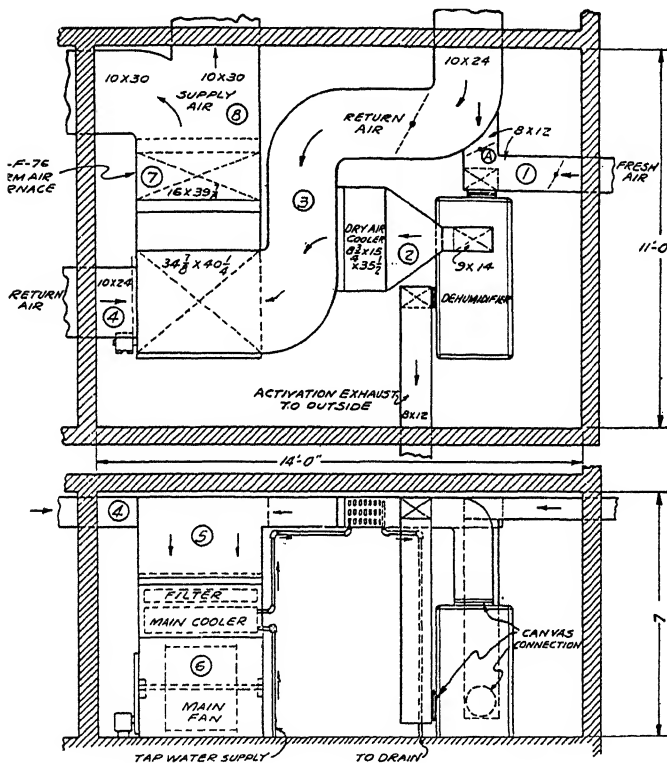
Fig. 211. Typical Layout of Dehumidifier with Main Fan
Courtesy of Bryant Heater Company, Cleveland, Ohio

return duct *3* to the dehumidifier, where it is dehumidified along with the outdoor air entering through duct *1*.

Cooling water enters the main cooler through the pipe shown, passes through the main cooler, and then passes on to the dry air cooler, from which it goes to the sewer.

Winter Operation. In the winter, the dampers in duct *1*, Fig. 211, and connection *A* are closed and the dehumidifier is inactive. Return air from the premises is conveyed to chamber *5* by return ducts *3* and

4, where it is filtered. A by-pass equipped with a damper is provided around the main cooler, the damper being in the open position. The fan in chamber *6* delivers the air to furnace casing *7*, where it is heated and humidified, and from which it then passes through supply duct *8* to the premises. The dampers located in ducts *1* and *2* are for the purpose of adjusting the proportion of recirculated air and outside air delivered to the dehumidifier.

Typical Dehumidifying Systems. Fig. 212 shows a dehumidifying system where the air is dehumidified separately and independently
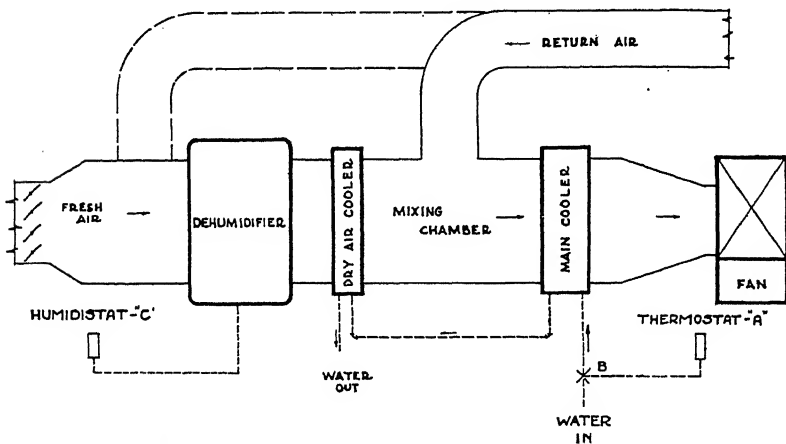
Fig. 212.   Silica Gel System Using Natural Water for Cooling
Courtesy of Bryant Heater Company, Cleveland, Ohio

of the cooling. Such a system is used in localities where water at 70°F. or below is available in ample amounts and is economical. The system provides for recirculation with and without dehumidifying, together with controlled amounts of outside air. From the study of effective temperatures, in another section, it can readily be seen that dehumidified air cooled by 68°F. to 70°F. water tends to make comfortable conditions.

In Fig. 212 thermostat *A* operates automatic water valve *B* on main cooler. The humidistat *C* automatically regulates the operation of the dehumidifier.

Fig. 213 shows a system employing silica gel dehumidifying with refrigeration equipment. This is representative of the system method used where water temperatures exceed 70°F. as previously mentioned.

In Fig. 213 the thermostat *A* controls the operation of the compressor.   The thermal expansion valve *B*, on evaporator, is for regulating the flow of liquid refrigerant to the evaporator and to maintain the proper refrigerant temperature.   The humidistat *C* automatically regulates the operation of the dehumidifier.   It should be remembered that in dehumidification the latent heat is removed by adsorption.
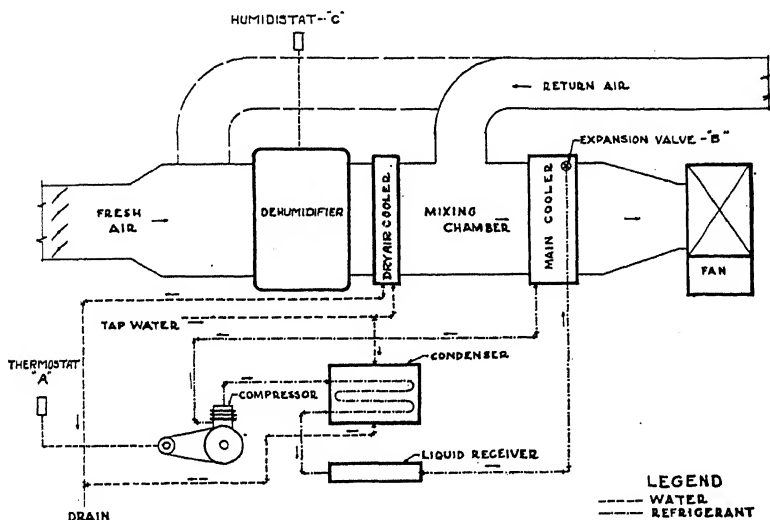
Fig. 213.   Silica Gel System Using Refrigeration for Cooling
Courtesy of Bryant Heater Company, Cleveland, Ohio

*Direct Expansion Coils.   Fig. 214 shows a direct expansion coil used for cooling.   Such coils have become widely used for cooling and air-conditioning applications which have brought about requirements for quick heat transfer from coil to air, regulated velocity through the coils, and a means of even distribution of gases through the coil for maximum efficiency.   Such requirements must be met before apparatus of this kind will function satisfactorily. Quick heat transfer refers to the speed of heat transfer from gas and depends upon the amount of cold surface with which the air can be brought in direct contact.

The handling of direct expansion gases through coils is exactly the same in principle as the handling of water.   With the latter a low velocity results in a stagnation on the inner surface of the tubes and

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*Data Courtesy of The Trane Company, LaCrosse, Wisconsin.

the same result is obtained in a direct expansion coil unless the velocities are properly proportioned to create a turbulence within the tube. For this reason a great deal of attention should be paid to the design of coil circuits to maintain sufficient velocities through each individual tube.

By tube sizing and circuit layout, proper distribution can be
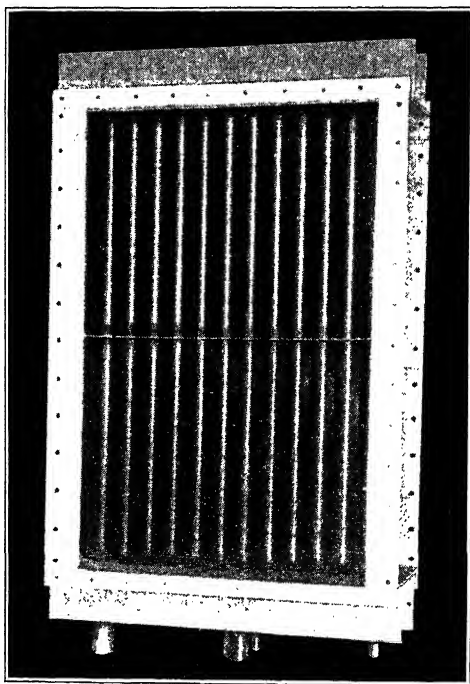
Fig. 214.   Direct Expansion Cooling Coil
Courtesy of The Trane Company, La Crosse, Wis.

assured.   Here circuiting makes a coil *live* over its entire surface and that is at once beneficial and necessary.   The even distribution of gas to each tube of the circuit is an extremely important factor in developing even temperatures over the faces of coils.

Cooling coils, as applied to air conditioning, using mechanical refrigerants as a cooling medium, encounter conditions so variable that to obtain maximum efficiency a coil must be designed for the condition under which it must operate.   First of importance is the

fact that the installation requirements may limit the actual dimen-
sions of the coil; second, the entering air temperature and air velocity
may cause extremely high or low loads per square foot of coil area;
and third, the properties of the refrigerant may place definite limita-
tions on the coil.

Studies of heat transfer have proved that the velocity of a cooling
medium through the coil greatly affects the rate of heat flow or
capacity of cooling coils. Should the velocity of the refrigerant be
low, the heat transfer will be low, due to stagnation of refrigerant.
Conversely, the refrigerant velocity may be so high that the rate of
heat transfer is not increased. This excessive velocity will cause a
high pressure drop over the evaporator coil, which decreases the
expansion valve and compressor capacity in addition to increasing
the super-heat in the coil to an objectionable degree.

A coil of a given size may have an extremely wide range of
capacities. However, if the design is not suitable for the extreme
conditions, the maximum capacity of the coil is not obtained and, in
addition, difficulties will be experienced with the expansion valve and
compressor, as previously explained.

Selecting Coils. The following information, while particularly
applicable to Trane coils, still gives a good explanation relative to
the selection of coils in general.

GENERAL NOTES ON CALCULATIONS

(1) The wet-bulb temperature indicates the total heat in the air, the sum
of both the sensible and latent.

(2) For air in a saturated condition the dry bulb, wet bulb, and dew point
are the same.

(3) The most economical range of air velocities over the face of a coil is
from 400 to 600 feet per minute.

(4) Air leaving a coil may be cooled to a condition of lower leaving dew
point than entering, and yet may be slightly unsaturated. This is due to the
varying temperatures of the coil surface, and in contact of the air with these various
temperatures, parts of it may be cooled to approximately the refrigerant temper-
ature, and others may not have the sensible heat entirely dissipated. The final
condition of the air will be unsaturated, yet the desired dew point is obtained.

For average entering air temperatures where the dew point is above the
refrigerant temperature, laboratory tests have shown that for a 2-row coil the
leaving relative humidity is about 85%, for a 4-row coil it is 90%, and for a 6-row
coil it is 93%.

(5) Most refrigerant cooling coil capacity ratings are based on flooded
operation with the refrigerant temperature above 32°F.

When selecting coils for direct expansion operation, it is necessary to make

provisions for the superheat of the leaving refrigerant gas that is necessary for both the proper operation of the expansion valve and condition of the refrigerant gas entering the compressor. This may be accomplished in the following three ways: (1) use of drier coil, as illustrated in Fig. 215; (2) addition of an extra row
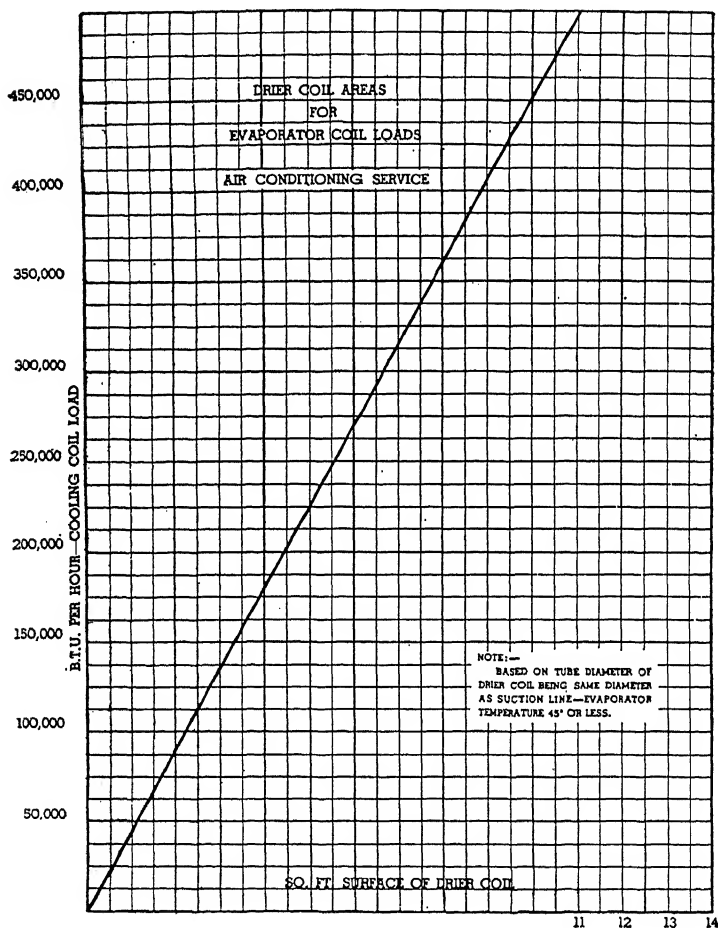
Fig. 215. Drier Coil Chart
Courtesy of The Trane Company, La Crosse, Wis.

of tubes to raise the gas temperature the required amount; and (3) allowance for superheat in coil calculation. Procedure for these methods is explained as follows:

Counter Flow and Parallel Flow of Air and Cooling Medium. Counter flow is obtained when the cooling medium enters the cooling section at the same end as the leaving air and leaves at the same end as the entering air. Counter flow should be used whenever possible because this means a higher mean temperature

difference so that less surface is required to do a given duty. Parallel flow is obtained when the air and water enter the cooling surfaces at the same end of the cooling bank.

Recommended Procedure. Given the following load conditions:

Air temperature entering coil—

82.25° dry bulb; 68° wet bulb; 60.9° dew point; 48% relative humidity.

Air temperature leaving coil—

58.6° saturated (see general notes on calculations—subject 2).

Air volume—

6650 c.f.m.

Coil Area Requirements. For purposes of calculation, an air velocity of 550 feet per minute is selected. (See general notes on calculations—subject 3.)

Since the coil area $= \dfrac{\text{c.f.m.}}{\text{face velocity}}$, the formula becomes $\dfrac{6,650}{500} = 13.3$ square feet of coil area required. Reference to Table 89 (page 381) shows that a $32 \times$ 60-inch coil has 13.3 square feet of face area.

Determination of Coil Load. The formula for determining the total load in B.t.u. per hour is:

$$(\text{pounds of air per hour}) \times (\text{total heat difference})$$

Reading the wet-bulb temperature of air entering the coils and the wet-bulb temperature of the air leaving the coils from Table 90 (page 382) shows that:

$$
\begin{aligned}
\text{The total heat at } 68° \quad \text{wet bulb} &= 31.92 \text{ B.t.u. per pound} \\
\text{The total heat at } 58.6° \text{ wet bulb} &= \underline{25.26 \text{ B.t.u. per pound}} \\
& 6.66 \text{ B.t.u. per pound}
\end{aligned}
$$

Note that the air leaving the coil is based on 58.5° saturated. (See general notes on calculations—subject 2.)

The next step is to reduce the air quantity of 6,650 c.f.m. to pounds of air per hour by the following formula:

$$\frac{6,650}{13.5^{*}} = 29,600 \text{ pounds per hour.}$$

According to the above formula, the total coil load then becomes:

$$29,600 \times 6.66 = 197,500 \text{ B.t.u. per hour, or the number of B.t.u.}$$
to be removed by the coil.

Determination of Mean Temperature Difference. (See general note on calculations—subject 5.) This coil may be used in conjunction with a drier coil, or an extra row of tubes added to accomplish the same results.

Counter flow of refrigerant and air is used on this coil, and the mean temperature difference is determined as follows:

1. Subtract the temperature of the leaving refrigerant from the temperature of the entering air.

2. Subtract the temperature of the entering refrigerant from the temperature of the leaving air. In other words:

*Cubic feet in one pound. See Psychrometric Chart in the back of the book.

Entering air.................................82.25
Leaving refrigerant..........................40.00
Greatest terminal difference.................42.25

Leaving air................................. 58.6
Entering refrigerant......................... 40.0
Least terminal difference.................... 18.6

Next, refer to Table 91 (page 383), where the terminal check shows a m.t.d. of 28.9°.

Note: This mean temperature difference is a logarithmic mean and not arithmetic.

Wet Surface Factor. It is commonly known that a wet surface produces greater heat transfer than a dry surface and the percentage of greater heat transfer is dependent upon the amount of moisture on the fins. This amount of moisture, in turn, is dependent upon the difference in temperature between the refrigerant and the dry bulb and dew point of the entering air.

To determine the increase in heat transfer due to this wet surface, it is necessary to know the difference between the entering dry-bulb temperature and refrigerant, and the difference between the entering dew point and refrigerant temperatures. In this discussion the differences, as explained, are as follows:

Difference between dry bulb and refrigerant:

82.25 dry bulb
40.00 refrigerant
42.25

Difference between dew point and refrigerant:

60.9 dew point
40.0 refrigerant
20.9

From the diagram, Fig. 216, the factor of increase can be calculated and found to be 1.29.

Determining Number of Rows of Tubes Required. To determine the number of rows, the actual coil load must be converted into a form which can be read from the capacity Table 92, page 384.

The load is converted as follows:

$$\frac{\text{Coil Load}}{\text{square feet (coil area)} \times \text{m.t.d.} \times} = \text{B.t.u. per hour per square foot face area per degree m.t.d.}$$

$$\text{f.w.} \begin{cases} \text{Wet surface} \\ \text{Factor change} \end{cases}$$

or

$$\frac{197{,}500}{13.3 \times 28.9 \times 1.29} = 399 \text{ B.t.u. per hour per square foot per degree m.t.d.}$$

Referring to Table 92, it is found that the nearest figure at 500 feet per minute face velocity is 464 B.t.u., or 4 rows of tubes.

Should it be desirable to allow for superheat of the refrigerant in the evaporator coil, one row of tubes must be added to the above selection of depth, making the final coil 32×60 inches 5-row coil, or a new calculation made allowing for superheat.

Drier Coil. The selection of the evaporator coil has been made on the basis of using a drier coil in the installation. To find the square feet of tube surface for this load, consult the diagram showing the surface area required for various
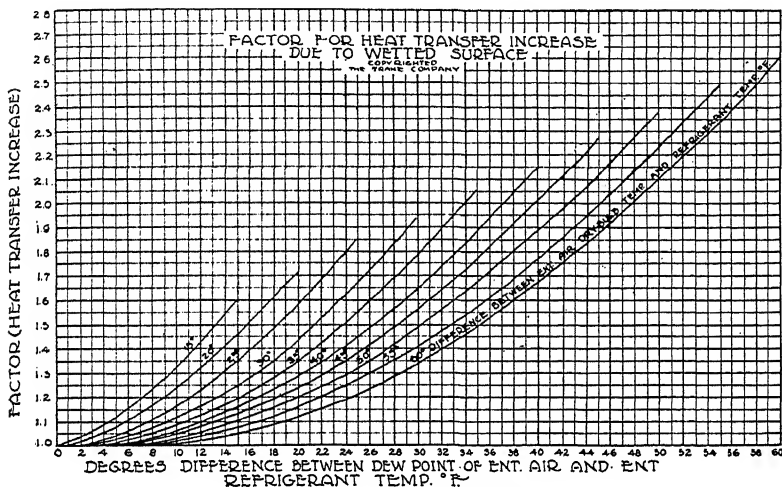
Fig. 216. Factor for Heat Transfer
Courtesy of The Trane Company, La Crosse, Wis.

loads in Fig. 215. From the diagram it can be calculated that 4.4 square feet of surface is required.

Note: When using diagrams, such as those represented in Figs. 215 and 216, extreme accuracy must be maintained. Interpolation is frequently necessary, and this also requires great accuracy.

Alternate Calculation. (Showing definite final air temperature and superheat.) Should it be desirable to figure the coil for a definite final air temperature, assume the leaving air as having 90% relative humidity. (See general notes on calculations—subject 4.) The final air condition may be determined from the Psychrometric Chart (found in the back of the book), as we know the leaving dew point is 58.6°. This reading shows leaving condition of:

61.6°F. dry bulb; 59.7°F. wet bulb;
58.6°F. dew point; 90% relative humidity.

Coil Load. *Total heat @ 68°F. wet bulb = 31.92 B.t.u.'s per pound
*Total heat @ 59.7°F. wet bulb = 25.98 B.t.u.'s per pound
———
5.94 B.t.u.'s per pound

B.t.u.'s per hour = 29,600×5.94 = 176,000.

*See Psychrometric Chart in the back of the book.

CHART 1

TYPES OF SERPENTINE FOR TRANE
DIRECT EXPANSION COOLING COILS
USING
METHYL CHLORIDE OR FREON REFRIGERANT

WHEN THE INTERSECTION OF LINE "A" FALLS BETWEEN THE SHADED AREAS DESIGNATING TYPES OF SERPENTINE, CHANGE FACE AREA DIMENSIONS OF COIL TO SECURE PROPER DESIGN.

IMPORTANT

WHERE POSSIBLE ADJUST THE RATIO OF COIL LENGTH TO WIDTH SO THAT STANDARD SERPENTINE MAY BE USED.

HALF STANDARD SERPENTINE

STANDARD SERPENTINE

DOUBLE STANDARD SERPENTINE

IF POINT "A" FALLS BELOW SHADED AREAS SEE CHART 2

EXAMPLE

GIVEN:
COIL LOAD 187,000 BTU HR.
COIL SIZE 32" X 60" 4 ROWS
REFRIGERANT—40° FREON
DETERMINE:
TYPE SERPENTINE TO BE USED
SOLUTION:
PROJECT LINE FROM LOAD SCALE AT LEFT THROUGH WIDTH SCALE TO LEFT HAND EDGE OF CHART RUN VERTICAL LINE DRAWN FROM LINE HORIZONTALLY TO INTERSECT LENGTH SCALE AT BOTTOM.

FREON
12"   16"   24"
                32"

METHYL CHLORIDE
12"   16"   24"
                32"

WIDTH OF COIL
IN INCHES

IF THE REFRIGERANT TO BE USED IS METHYL CHLORIDE USE THE SCALE ON THE LEFT. IF FREON IS USED—USE RIGHT HAND SCALE.

2 ROW

3 ROW

4 ROW

5 ROW

6 ROW

COIL LENGTH IN INCHES

12 18 24 30 36 42 48 60 72 84 96 108 120

Fig. 217. Alignment Chart
Courtesy of The Trane Company, La Crosse, Wis.

COIL LOAD IN B.T.U. PER HR.

15,000
20,000
30,000
40,000
50,000
60,000
70,000
80,000
90,000
100,000
150,000
200,000
300,000
400,000
500,000
600,000
700,000
800,000
900,000
1,000,000

Mean Temperature Difference.

$$
\begin{array}{r}
82.25 \text{ entering air} \\
\text{Refrigerant } 45.00 \\
\hline
37.25 \text{ greatest terminal difference}
\end{array}
$$

$$
\begin{array}{r}
61.6 \text{ leaving air} \\
\text{Refrigerant } 40.0 \\
\hline
21.6 \text{ least terminal difference}
\end{array}
$$

The mean temperature difference from Table 91 (page 383) is 28.8. Wet surface factor is 1.23 (same as previously used).

Number of rows of tubes:

$$
\frac{176,000}{13.3 \times 28.8 \times 1.29} = 356
$$

From Table 92 (page 384), we find that three rows of tubes at 500 f.p.m. will give 348 B.t.u.'s, so that a 4-row coil must be used. The alternative of using another row of tubes or a drier coil for the required superhead for this latter selection of evaporator coil remains and may be treated as previously explained.

Selection of a Serpentine Design Coil. The method of determining coil design has been worked out as an illustrative example on the alignment chart, Fig. 217, using the total load of 197,500 B.t.u. per hour and coil size 32×60 inches 4 rows deep with Freon refrigerant.

Table 89. *Square Feet Face Area of Trane Direct Expansion Cooling Coils for Various Sizes

Coil Width or Ht. in In.	Coil Length—Inches														
	12	15	18	24	30	36	42	48	54	60	66	72	78	84	90
12	1.0	1.25	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5
16	...	1.65	2.0	2.66	3.33	4.0	4.66	5.33	6.0	6.66	7.33	8.0	8.66	9.3	10
24	...	....	....	4.0	5.0	6.0	7.0	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15
32	...	....	....	....	....	8.0	9.33	10.66	12.0	13.33	14.66	16.0	17.33	18.66	20

*Courtesy of The Trane Company, La Crosse, Wis.

Table 90. *Total Heat Content of Air at Various Wet=Bulb Temperatures

Interpolated to Tenths of a Degree from the Late Professor Goodenough's Table of "Properties of Air"

Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.	Wet Bulb Tem.	B.t.u. per Lb.
Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9	Col. 10	Col. 11	Col. 12	Col. 13	Col. 14	Col. 15	Col. 16
40.0	15.21	45.0	17.59	50.0	20.19	55.0	23.04	60.0	26.18	65.0	29.65	70.0	33.51	75.0	37.81
.1	15.26	.1	17.64	.1	20.25	.1	23.10	.1	26.25	.1	29.72	.1	33.59	.1	37.90
.2	15.30	.2	17.69	.2	20.30	.2	23.16	.2	26.31	.2	29.80	.2	33.67	.2	37.99
.3	15.35	.3	17.74	.3	20.36	.3	23.22	.3	26.38	.3	29.87	.3	33.76	.3	38.09
.4	15.39	.4	17.79	.4	20.41	.4	23.28	.4	26.44	.4	29.95	.4	33.84	.4	38.18
40.5	15.44	45.5	17.84	50.5	20.47	55.5	23.34	60.5	26.51	65.5	30.02	70.5	33.92	75.5	38.27
.6	15.49	.6	17.89	.6	20.52	.6	23.40	.6	26.58	.6	30.09	.6	34.00	.6	38.36
.7	15.53	.7	17.94	.7	20.58	.7	23.46	.7	26.64	.7	30.17	.7	34.08	.7	38.45
.8	15.58	.8	17.99	.8	20.63	.8	23.52	.8	26.71	.8	30.24	.8	34.17	.8	38.55
.9	15.62	.9	18.04	.9	20.69	.9	23.58	.9	26.77	.9	30.32	.9	34.25	.9	38.64
41.0	15.67	46.0	18.09	51.0	20.74	56.0	23.64	61.0	26.84	66.0	30.39	71.0	34.33	76.0	38.73
.1	15.72	.1	18.14	.1	20.80	.1	23.70	.1	26.91	.1	30.47	.1	34.41	.1	38.82
.2	15.76	.2	18.19	.2	20.85	.2	23.76	.2	26.98	.2	30.54	.2	34.50	.2	38.92
.3	15.81	.3	18.24	.3	20.91	.3	23.82	.3	27.04	.3	30.62	.3	34.58	.3	39.01
.4	15.86	.4	18.29	.4	20.96	.4	23.88	.4	27.11	.4	30.69	.4	34.67	.4	39.11
41.5	15.91	46.5	18.35	51.5	21.02	56.5	23.95	61.5	27.18	66.5	30.77	71.5	34.75	76.5	39.20
.6	15.95	.6	18.40	.6	21.08	.6	24.01	.6	27.25	.6	30.85	.6	34.83	6	39.29
.7	16.00	.7	18.45	.7	21.13	.7	24.07	.7	27.32	.7	30.92	.7	34.92	.7	39.39
.8	16.05	.8	18.50	.8	21.19	.8	24.13	.8	27.38	.8	31.00	.8	35.00	.8	39.48
.9	16.09	.9	18.55	.9	21.24	.9	24.19	.9	27.45	.9	31.07	.9	35.09	.9	39.58
42.0	16.14	47.0	18.60	52.0	21.30	57.0	24.25	62.0	27.52	67.0	31.15	72.0	35.17	77.0	39.67
.1	16.19	.1	18.65	.1	21.36	.1	24.31	.1	27.59	.1	31.23	.1	35.26	.1	39.77
.2	16.24	.2	18.70	.2	21.41	.2	24.38	.2	27.66	.2	31.30	.2	35.34	.2	39.86
.3	16.28	.3	18.76	.3	21.47	.3	24.44	.3	27.73	.3	31.38	.3	35.43	.3	39.96
.4	16.33	.4	18.81	.4	21.53	.4	24.50	.4	27.80	.4	31.46	.4	35.51	.4	40.06
42.5	16.38	47.5	18.86	52.5	21.59	57.5	24.57	62.5	27.87	67.5	31.54	72.5	35.60	77.5	40.16
.6	16.43	.6	18.91	.6	21.64	.6	24.63	.6	27.94	.6	31.61	.6	35.69	.6	40.25
.7	16.48	.7	18.96	.7	21.70	.7	24.69	.7	28.01	.7	31.69	.7	35.77	.7	40.35
.8	16.52	.8	19.02	.8	21.76	.8	24.75	.8	28.08	.8	31.77	.8	35.86	.8	40.45
.9	16.57	.9	19.07	.9	21.81	.9	24.82	.9	28.15	.9	31.84	.9	35.94	.9	40.54
43.0	16.62	48.0	19.12	53.0	21.87	58.0	24.88	63.0	28.22	68.0	31.92	73.0	36.03	78.0	40.64
.1	16.67	.1	19.17	.1	21.93	.1	24.94	.1	28.29	.1	32.00	.1	36.12	.1	40.74
.2	16.72	.2	19.23	.2	21.99	.2	25.01	.2	28.36	.2	32.08	.2	36.21	.2	40.84
.3	16.76	.3	19.28	.3	22.04	.3	25.07	.3	28.43	.3	32.16	.3	36.29	.3	40.94
.4	16.81	.4	19.33	.4	22.10	.4	25.14	.4	28.50	.4	32.24	.4	36.38	.4	41.04
43.5	16.86	48.5	19.39	53.5	22.16	58.5	25.20	63.5	28.58	68.5	32.32	73.5	36.47	78.5	41.14
.6	16.91	.6	19.44	.6	22.22	.6	25.26	.6	28.65	.6	32.39	.6	36.56	.6	41.23
.7	16.96	.7	19.49	.7	22.28	.7	25.33	.7	28.72	.7	32.47	.7	36.65	.7	41.33
.8	17.00	.8	19.55	.8	22.33	.8	25.39	.8	28.79	.8	32.55	.8	36.73	.8	41.43
.9	17.05	.9	19.60	.9	22.39	.9	25.46	.9	28.86	.9	32.63	.9	36.82	.9	41.53
44.0	17.10	49.0	19.65	54.0	22.45	59.0	25.52	64.0	28.93	69.0	32.71	74.0	36.91	79.0	41.63
.1	17.15	.1	19.70	.1	22.51	.1	25.59	.1	29.00	.1	32.79	.1	37.00	.1	41.73
.2	17.20	.2	19.76	.2	22.57	.2	25.65	.2	29.07	.2	32.87	.2	37.09	.2	41.83
.3	17.25	.3	19.81	.3	22.63	.3	25.72	.3	29.15	.3	32.95	.3	37.18	.3	41.93
.4	17.30	.4	19.87	.4	22.69	.4	25.78	.4	29.22	.4	33.03	.4	37.27	.4	42.03
44.5	17.35	49.5	19.92	54.5	22.75	59.5	25.85	64.5	29.29	69.5	33.11	74.5	37.36	79.5	42.14
.6	17.39	.6	19.97	.6	22.80	.6	25.92	.6	29.36	.6	33.19	.6	37.45	.6	42.24
.7	17.44	.7	20.03	.7	22.86	.7	25.98	.7	29.43	.7	33.27	.7	37.54	.7	42.34
.8	17.49	.8	20.08	.8	22.92	.8	26.05	.8	29.51	.8	33.35	.8	37.63	.8	42.44
.9	17.54	.9	20.14	.9	22.98	.9	26.11	.9	29.58	.9	33.43	.9	37.72	.9	42.54

*Courtesy of The Trane Company, La Crosse, Wisconsin.

Table 9 *Giving Log Mean Temperature Difference

Least or Greatest Terminal Temperature Difference

	1	2	3	4	5	6	7	8	9	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44
1	1.00	1.44	1.82	2.16	2.49	2.79	3.08	3.37	3.64	3.91	4.43	4.93	5.41	5.88	6.34	6.79	7.24	7.67	8.10	8.53	8.94	9.36	9.76	10.17	10.57	10.97	11.36
2	1.44	2.00	2.47	2.89	3.27	3.64	3.99	4.33	4.65	4.97	5.58	6.17	6.73	7.28	7.82	8.34	8.85	9.36	9.85	10.34	10.82	11.29	11.76	12.22	12.69	13.14	13.59
3	1.82	2.47	3.00	3.48	3.92	4.33	4.72	5.10	5.46	5.81	6.49	7.14	7.77	8.37	8.96	9.54	10.10	10.65	11.19	11.73	12.24	12.77	13.28	13.79	14.28	14.78	15.27
4	2.16	2.89	3.48	4.00	4.48	4.93	5.36	5.77	6.17	6.55	7.28	7.98	8.66	9.31	9.94	10.56	11.16	11.76	12.33	12.90	13.47	14.02	14.56	15.10	15.63	16.16	16.68
5	2.49	3.27	3.92	4.48	5.00	5.49	5.94	6.38	6.81	7.21	8.00	8.74	9.45	10.14	10.82	11.48	12.11	12.74	13.35	13.95	14.55	15.13	15.71	16.27	16.83	17.39	17.94
6	2.79	3.64	4.33	4.93	5.49	6.00	6.49	6.95	7.40	7.83	8.66	9.44	10.20	10.92	11.63	12.31	12.98	13.64	14.28	14.91	15.53	16.14	16.74	17.34	17.92	18.50	19.07
7	3.08	3.99	4.72	5.36	5.94	6.49	7.00	7.49	7.96	8.41	9.28	10.10	10.89	11.65	12.38	13.10	13.80	14.48	15.15	15.80	16.45	17.09	17.71	18.33	18.93	19.53	20.13
8	3.37	4.33	5.10	5.77	6.38	6.95	7.49	8.00	8.49	8.96	9.87	10.72	11.54	12.33	13.10	13.84	14.56	15.27	15.96	16.64	17.31	17.97	18.62	19.25	19.88	20.50	21.12
9	3.64	4.65	5.46	6.17	6.81	7.40	7.96	8.49	9.00	9.49	10.43	11.32	12.17	12.98	13.78	14.55	15.29	16.02	16.74	17.44	18.13	18.81	19.48	20.13	20.78	21.42	22.05
10	3.91	4.97	5.81	6.55	7.21	7.83	8.41	8.96	9.49	10.00	10.97	11.89	12.77	13.61	14.43	15.22	15.99	16.74	17.47	18.21	18.91	19.61	20.30	20.98	21.64	22.30	22.94
12	4.43	5.58	6.49	7.28	8.00	8.66	9.28	9.87	10.43	10.97	12.00	12.97	13.90	14.79	15.66	16.50	17.31	18.11	18.88	19.64	20.39	21.12	21.85	22.56	23.26	23.95	24.63
14	4.93	6.17	7.14	7.98	8.74	9.44	10.10	10.72	11.32	11.89	12.97	14.00	14.98	15.92	16.82	17.70	18.55	19.39	20.20	20.99	21.77	22.54	23.29	24.04	24.77	25.49	26.20
16	5.41	6.73	7.77	8.66	9.45	10.20	10.89	11.54	12.17	12.77	13.90	14.98	16.00	16.98	17.93	18.84	19.73	20.60	21.44	22.27	23.08	23.88	24.66	25.44	26.19	26.94	27.68
18	5.88	7.28	8.37	9.31	10.14	10.92	11.65	12.33	12.98	13.61	14.79	15.92	16.98	18.00	18.98	19.93	20.86	21.76	22.64	23.49	24.33	25.16	25.97	26.77	27.55	28.33	29.09
20	6.34	7.82	8.96	9.94	10.82	11.63	12.38	13.10	13.78	14.43	15.66	16.82	17.93	18.98	20.00	20.98	21.94	22.88	23.78	24.66	25.53	26.39	27.22	28.05	28.85	29.65	30.44
22	6.79	8.34	9.54	10.56	11.48	12.31	13.10	13.84	14.55	15.22	16.50	17.70	18.84	19.93	20.98	22.00	22.98	23.94	24.88	25.79	26.69	27.57	28.43	29.28	30.11	30.93	31.74
24	7.24	8.85	10.10	11.16	12.11	12.98	13.80	14.56	15.29	15.99	17.31	18.55	19.73	20.86	21.94	22.98	24.00	24.98	25.94	26.89	27.81	28.71	29.59	30.47	31.32	32.16	33.00
26	7.67	9.36	10.65	11.76	12.74	13.64	14.48	15.27	16.02	16.74	18.11	19.39	20.60	21.76	22.88	23.94	24.98	26.00	26.98	27.95	28.90	29.82	30.73	31.62	32.50	33.36	34.21
28	8.10	9.85	11.19	12.33	13.35	14.28	15.15	15.96	16.74	17.47	18.88	20.20	21.44	22.64	23.78	24.88	25.94	26.98	28.00	28.98	29.96	30.90	31.83	32.74	33.64	34.52	35.40
30	8.53	10.34	11.73	12.90	13.95	14.91	15.80	16.64	17.44	18.21	19.64	21.00	22.27	23.49	24.66	25.79	26.89	27.95	28.98	30.00	31.00	31.95	32.91	33.84	34.76	35.66	36.55
32	8.94	10.82	12.24	13.47	14.55	15.53	16.45	17.31	18.13	18.91	20.39	21.77	23.08	24.33	25.53	26.69	27.81	28.90	29.96	31.00	32.00	32.98	33.96	34.90	35.86	36.78	37.68
34	9.36	11.29	12.77	14.02	15.13	16.14	17.09	17.97	18.82	19.61	21.12	22.54	23.88	25.16	26.39	27.57	28.71	29.82	30.90	31.95	32.98	34.00	34.97	35.97	36.92	37.86	38.79
36	9.76	11.76	13.28	14.56	15.71	16.74	17.71	18.62	19.48	20.30	21.85	23.29	24.66	25.97	27.22	28.43	29.59	30.73	31.83	32.91	33.96	34.97	36.00	36.97	37.95	38.91	39.86
38	10.17	12.22	13.79	15.10	16.27	17.34	18.33	19.25	20.13	20.98	22.56	24.04	25.44	26.77	28.05	29.28	30.47	31.62	32.74	33.84	34.90	35.97	36.97	38.00	38.99	39.96	40.93
40	10.57	12.69	14.28	15.63	16.83	17.92	18.93	19.88	20.78	21.64	23.26	24.77	26.19	27.55	28.85	30.11	31.32	32.50	33.64	34.76	35.86	36.92	37.95	38.99	40.00	40.98	41.97
44	11.36	13.59	15.27	16.68	17.93	19.07	20.13	21.12	22.05	22.95	24.63	26.20	27.68	29.09	30.44	31.74	33.00	34.21	35.40	36.55	37.68	38.79	39.86	40.93	41.97	43.01	44.00
48	12.14	14.47	16.23	17.71	19.01	20.20	21.30	22.32	23.30	24.23	25.97	27.60	29.13	30.59	31.98	33.32	34.63	35.88	37.11	38.30	39.46	40.60	41.71	42.81	43.88	44.94	45.97
50	12.53	14.91	16.71	18.21	19.54	20.75	21.87	22.92	23.91	24.85	26.63	28.28	29.84	31.32	32.74	34.10	35.42	36.70	37.94	39.15	40.33	41.48	42.62	43.73	44.82	45.88	46.94
52	12.91	15.35	17.18	18.71	20.07	21.30	22.44	23.51	24.51	25.47	27.28	28.96	30.54	32.05	33.49	34.87	36.21	37.51	38.77	40.00	41.19	42.36	43.51	44.64	45.73	46.82	47.88
54	13.29	15.78	17.64	19.21	20.59	21.85	23.00	24.09	25.11	26.09	27.92	29.63	31.24	32.77	34.23	35.64	36.99	38.31	39.59	40.83	42.05	43.23	44.39	45.53	46.65	47.75	48.83
56	13.66	16.21	18.11	19.70	21.11	22.39	23.56	24.67	25.71	26.70	28.56	30.30	31.93	33.48	34.96	36.39	37.77	39.10	40.40	41.65	42.89	44.09	45.27	46.42	47.55	48.67	49.75
58	14.04	16.63	18.57	20.19	21.62	22.92	24.12	25.24	26.30	27.31	29.20	30.96	32.61	34.18	35.69	37.14	38.53	39.89	41.20	42.48	43.72	44.93	46.13	47.29	48.44	49.57	50.69
60	14.41	17.05	19.03	20.68	22.13	23.45	24.67	25.81	26.88	27.90	29.82	31.61	33.29	34.88	36.41	37.87	39.29	40.66	41.99	43.28	44.54	45.77	46.98	48.16	49.32	50.46	51.58
62	14.78	17.47	19.48	21.16	22.64	23.98	25.21	26.37	27.46	28.50	30.45	32.26	33.96	35.58	37.12	38.61	40.04	41.43	42.77	44.07	45.36	46.61	47.83	49.03	50.19	51.35	52.49
64	15.15	17.89	19.93	21.64	23.14	24.50	25.76	26.93	28.04	29.09	31.06	32.90	34.62	36.26	37.83	39.33	40.78	42.18	43.55	44.87	46.17	47.43	48.66	49.88	51.06	52.23	53.38
66	15.51	18.30	20.38	22.12	23.64	25.02	26.29	27.49	28.61	29.67	31.68	33.54	35.28	36.94	38.53	40.05	41.52	42.94	44.32	45.66	46.96	48.24	49.50	50.72	51.92	53.10	54.26
68	15.88	18.72	20.83	22.59	24.14	25.54	26.83	28.04	29.17	30.26	32.28	34.17	35.94	37.62	39.22	40.76	42.25	43.69	45.08	46.44	47.76	49.05	50.31	51.56	52.77	53.96	55.14
70	16.24	19.13	21.27	23.06	24.63	26.05	27.36	28.58	29.74	30.83	32.89	34.79	36.59	38.29	39.91	41.47	42.98	44.43	45.84	47.21	48.54	49.85	51.14	52.39	53.61	54.82	55.99
72	16.60	19.53	21.71	23.53	25.12	26.56	27.89	29.13	30.30	31.41	33.49	35.42	37.23	38.95	40.60	42.17	43.69	45.16	46.59	47.97	49.33	50.65	51.94	53.20	54.44	55.66	56.85

Least or Greatest Terminal Temperature Difference.

Table 92. Capacities in B.t.u. with Refrigerants Above 32 Degrees

Table showing capacity of Trane Direct Expansion Surface in B.t.u. per Hour per Degree Mean Temperature Difference per Square Foot Face Area. Capacities in this table are for direct expansion refrigerants above 32°F. and based on approximately 2200 feet per minute gas velocity.

No. of Rows of Tubes	Face Velocity in Feet per Minute. Air at 70°F.								
	200	300	400	500	600	700	800	900	1000
1	80	101	110	116	125	126	128	131	133
2	161	202	220	232	250	252	256	261	265
3	241	303	330	348	375	378	384	392	398
4	323	404	440	464	500	504	512	522	530
5	403	505	550	580	625	630	640	653	663
6	483	606	660	696	750	756	768	783	795
8	644	808	880	928	1000	1008	1024	1044	1060
10	805	1010	1100	1160	1250	1260	1280	1305	1325
12	966	1212	1320	1392	1500	1512	1536	1566	1590

Selecting Expansion Valves. Direct expansion coils should be carefully selected on the following basis to secure proper performance:

(1) Valve capacity should be 35% greater than actual coil load.

(2) Valve should be selected to operate against a back pressure 10 pounds per square inch higher than actual evaporator pressure.

(3) In determining head pressure on valves, allowance must be made for loss of pressure in liquid line due to friction and difference in elevation between condenser and evaporator.

(4) In determining the pressure difference across the expansion valve, be sure the lowest condenser pressure which may be encountered during the cooling season is used.

Air Friction through Coils (Dry). For the type of coil explained in this section, Table 93 gives air friction in inches of water through direct expansion coils. The table assumes the coil is dry.

Table 93. Table of Air Friction in Inches of Water through Trane Direct Expansion Coils (Dry Coil)

No. of Rows of Tubes	Face Velocity in Feet per Minute							
	200	300	400	500	600	700	800	1000
1	.018	.032	.055	.080	.110	.145	.185	.27
2	.025	.048	.080	.120	.155	.210	.260	.39
3	.032	.060	.100	.153	.210	.270	.335	.51
4	.040	.075	.125	.190	.257	.332	.410	.63
6	.057	.100	.173	.263	.352	.460	.560	.87
8	.072	.130	.220	.338	.450	.587	.710	1.11
10	.088	.155	.270	.410	.550	.715	.860	1.35
12	.102	.180	.315	.482	.650	.840	1.10	1.60
14	.120	.207	.365	.555	.745	.965	1.17	1.84

Air friction based on Standard Fin Spacing. For air friction using 3 fins per inch, multiply above factors by 0.70; if 2 fins per inch, multiply by 0.55. (Standard fin spacing is 6½ fins per inch.)

Heating Coils. Fan systems of winter air conditioning, except some of the furnace types, require heaters of the coil type to warm the air. The number of coils, installed depends on the temperature rise through which the air is to be warmed. This same principle applies to all heating coils and the rate of heat transmission varies almost as the difference in temperature between the steam and the air. Thus the first unit of coils the air contacts has a greater transmission than the second set, etc. The number of units used in any
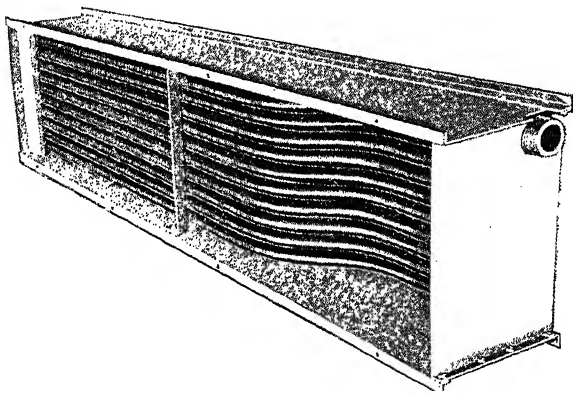
Fig. 218. Typical Unit of Aerofin
Courtesy of Aerofin Corporation, Newark, N. J.

given case depends on temperature requirements and outdoor conditions. Thus the final temperature depends on the number of units and also upon the velocity of the air. To illustrate heating coils and their selection the following discussion and typical examples are given:

Aerofin. Aerofin was designed by experienced fan engineers to meet, scientifically and specifically, every requirement of modern, advanced fan system practice and to provide a heat-surface so readily adaptable that it would inspire, as well as permit, a constant advancement and improvement in such practice. Fig. 218 illustrates a typical Aerofin.

The designers of Aerofin set up as prerequisite seven specifications:

    (1) Non-corrodible, ductile (i.e. not brittle) material
    (2) Material having high rate of thermal conductivity
    (3) Light-weight
    (4) Compactness
    (5) Sturdiness
    (6) Standardized encased unit design
    (7) Arrangement affording simple, quick, economical installation

For the accomplishment of this ideal the metal best adapted is copper. Copper is ductile, therefore not subject to breakage in shipment or handling. It is non-corrodible and, in ordinary service, substantially imperishable. It is so strong and of such uniform structure that drawn seamless tubes having relatively thin walls possess ample strength. Copper therefore permits a design which results in a heat-surface astonishingly light in weight, durable, and sturdy.

Thermal conductivity is closely related to electrical conductivity and the same characteristics which make copper so indispensable for electrical purposes make it equally desirable for the transmission of heat. In fact, copper transmits heat some eight times as rapidly as cast iron; the final reason for its selection as the material of which the heat-surface of Aerofin is constructed.

With so adequate a material as a basis the next step in the design was a careful study of the potentialities of extended surface. It was determined that a properly designed extended surface made of highly conductive copper, *if metallically bonded to the primary surface,* is fully effective in the ratio of four to one. That is, it was found feasible, with proper design, to employ 80% extended surface and but 20% primary surface. This is the approximate ratio in Aerofin.

The bonding of the extended surface to the primary or tube surface is achieved by the use of a special high-temperature solder. This method was found superior to all others, since it is a true *metallic* bond which affords an uninterrupted transmission of heat, as effectively as though the primary and extended surface were actually integral.

Again the analogy between thermal and electrical conductivity, —all wiring splices are soldered, to insure unimpaired and permanent conductivity.

The *Aerofin tables,** as given in the text, in most cases have been abbreviated. Enough of each table is given to allow a typical example to be worked out. The reader can secure complete tables from the manufacturer at any time.

Example 1. It is required to heat 12,450 cubic feet of air per minute, from 0°F. to either 85.1°F. or 112°F. using steam pressure of 5 pounds and assuming a face velocity of 500 feet per minute (equivalent to 1000 feet per minute free area velocity). Select the required heating coils.

*These tables are prepared for illustrative purposes. The manufacturers' tables are used exactly as explained here.

Solution. From Table 94 for 5 pounds of steam it is shown that one unit deep will give a temperature rise from 0°F. to 31.8°F., and that the friction will be 0.061 inches of water (Table 95). This is the outside tempering coil (shown at *A* in Figs. 41, 42, and 43 in Chapter VI) handling the air below freezing temperature, and should be controlled from the fresh air intake (also shown by Figs. 41, 42, and 43 in Chapter VI) with a positive acting thermostat or else hand controlled.

The net face area required will be 12,450 ÷ 500 = 24.9 square feet.

From Table 96, it is shown that two units wide of Aerofin 61, having tubes 6 feet long, will give 24.9 square feet net face area. These may be set two units high one above the other, tubes horizontal, or two units wide, one beside the other, tubes vertical.

Table 96 also shows the over-all dimensions of the battery to be 4 feet 10 inches wide by 6 feet 8½ inches high, if set two units wide, tubes vertical; or 6 feet 8½ inches wide by 4 feet 10 inches high, if set two units high, tubes horizontal. In either case the depth of the heater is 10 inches (standard depth of casing in direction of flow) and the weight is 278 pounds.

For a re-heating coil, selection can be made of Aerofin 81, Table 97, two units wide as indicated in Table 98. This reheater will heat the air (at 500 feet face velocity and 5 pounds steam pressure) from *30°F. to 83.8°F. or from 31.8°F. to 85.1°F.* with a friction loss of 0.083 inches of water (Table 95).

To determine the italic figures, which refer to the different temperatures in the preceding paragraphs, Table 99 is used. In Table 97 it is seen that where the entering air temperature is +30°F. and at 500 feet velocity the final temperature is 83.8°F. However, the entering air is 31.8°F. because that is the temperature at which it left the first or tempering coil. The directions at the top of Table 99 tell how to calculate the final temperature when the entering air is 31.8°F. instead of 30°F.

The conversion factor for entering air (5 pounds steam) of 30°F. is .868. The factor for entering air at 40°F. is .824. This requires interpolation. There are 10 degrees between 30°F. and 40°F. Then to find the factor for 31.8°F. proceed as follows:

$$.868 - .824 = .044$$
$$.044 \div 10 = .0044$$
$$.0044 \times .8 = .00352$$
$$.0044 + .00352 = .00792$$
$$.868 - .00792 = .86008$$
$$.860 \times 62 = 53.320$$
$$53.3 + 31.8 = 85.1°F.$$

Thus the final temperature of 85.1°F. is established.

The weight of the two units will be 354 pounds.

As an alternate, one can select Aerofin 72, Table 100, and two units wide as shown in Table 101. This reheater will heat this air volume (500 feet) from 30°F. to 111°F. or from 31.8°F. to 112°F., with a friction of .116 inches of water. Interpolation is necessary as before. See Table 95 for friction. The total weight of these two units will be 456 pounds.

It will be noted that if the combination of Aerofin 61 and 81 is used under the above conditions, the temperature rise will be from 0°F. to 85.1°F. and the combined friction will be 0.144 inches of water.

If the combination of Aerofin 61 and 72 is used, the temperature rise will be from 0°F. to 112°F. and the combined friction 0.177 inches of water.

If higher air temperatures are desired, use additional sections of either Aerofin 71 and 72 or 81 and 82.

Table 94. Final Temperatures and Condensations
5 lbs. Steam—227°F. Temperature
FLEXITUBE AEROFIN-61

Temp. Entering Air	Units Deep	Velocity of Air through Net Face Area*—in feet per minute—Measured at 70°F. and 29.92″ Barometer							
		300 Ft. Face Velocity		400 Ft. Face Velocity		500 Ft. Face Velocity		600 Ft. Face Velocity	
		Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft.H.S.†	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.
−30°	1	19.2	1.78	14.2	2.14	10.7	2.46	8.2	2.76
	2	64.8	1.72	56.4	2.08	50.4	2.42	45.0	2.71
−20°	1	27.3	1.71	22.5	2.05	19.2	2.36	16.8	2.66
	2	69.2	1.66	61.2	1.96	55.4	2.27	50.9	2.56
−10°	1	35.4	1.64	30.8	1.97	27.5	2.26	25.2	2.54
	2	73.6	1.51	65.4	1.82	59.8	2.10	56.3	2.40
0°	1	43.5	1.57	39.1	1.88	31.8°	2.17	33.8	2.44
	2	78.9	1.43	71.0	1.71	64.8°	1.98	62.6	2.26
+10°	1	51.6	1.50	47.4	1.80	44.4	2.07	42.3	2.34
+20°	1	59.6	1.43	55.6	1.72	52.8	1.98	50.8	2.23
+30°	1	67.8	1.37	63.9	1.64	61.2	1.88	59.3	2.12
+40°	1	75.8	1.29	72.2	1.56	69.6	1.78	67.8	2.01
+50°	1	83.9	1.23	80.5	1.47	78.1	1.70	76.4	1.91

*Net Face Area means only that area facing the tubes and does not include the headers or ngs.
†Abbreviation H.S. stands for Heating Surface.

Note: Flexitube Aerofin-61 is furnished in one-row Units *only*, for Tempering service. Where Entering Air temperature is not lower than 0°F, one Unit should be used. Where Entering Air temperature is lower than 0°F, two Units should be used. All these Units should have Steam Supply controlled from Entering Air intake.

In all Flexitube Aerofin designations (such as Flexitube Aerofin-61 or Flexitube Aerofin-72) the numerals following the name indicate the *type of tubing* (first numeral) and the *number of rows of tubes deep* (in direction of air flow), (second numeral). Thus Flexitube Aerofin-61 designates tubing with ¼ inch fins, pitched 6 per inch, and a Unit 1 row of tubes deep; while Flexitube Aerofin-72 designates tubing with ⅜ inch fins, pitched 7 per inch, and a Unit 2 rows of tubes deep.

Air Velocity and Friction. By using the total net *face* area of Aerofin, instead of using free area, as with cast iron, engineering calculations are based on approximately *half* the old familiar velocities, because the net *face* area of Aerofin (only that area facing tubes, not including headers or casing) is approximately twice the free area, and thus the actual velocity through Aerofin is the same as that through equivalent cast iron.

Therefore a face velocity of 600 feet per minute through Aerofin corresponds to a free area velocity of approximately 1200 feet per minute through cast iron.

The friction through Aerofin is about the same as that through equivalent cast iron surface.

For public building or factory work the preferable velocities are from 500 to 800 feet per minute through net face area.

Table 95. Friction of Air—In Inches of Water—
For 70°F. and 29.92″ Barometer*

Rows of Helical Tubing Deep	Velocity of Air in Feet per Minute through Net Face Area†					
	300	400	500	600	700	800
FLEXITUBE AEROFIN-61						
1	.020	.037	.061	.090	.127	.170
2	.038	.068	.106	.154	.210	.276
FLEXITUBE AEROFIN-71 and 72						
1	.030	.050	.074	.103	.136	.174
2	.044	.076	.116	.164	.221	.285
3	.058	.101	.158	.225	.306	.396
4	.072	.127	.200	.286	.391	.507
5	.087	.153	.241	.347	.476	.618
6	.101	.178	.283	.408	.561	.729
7	.115	.204	.325	.469	.646	.840
8	.130	.230	.367	.530	.731	.951
FLEXITUBE AEROFIN-81 and 82						
1	.032	.055	.083	.117	.157	.200
2	.055	.095	.145	.205	.275	.353
3	.078	.135	.207	.293	.393	.506
4	.101	.175	.269	.381	.511	.659
5	.124	.215	.331	.469	.629	.812
6	.147	.255	.393	.557	.747	.965
7	.170	.295	.455	.645	.865	1.118

*For temperatures other than 70°F., refer to data which can be found in manufacturers' catalogue.
†Net Face Area means only that area facing the tubes and does not include headers or casing.

Table 96. Flexitube Aerofin=61—Physical Data

12-Tube Face—20 9/16-Inch Casing

Length of Tubes Feet	Net Face Area* Sq. Ft.	Inside Dimensions Over Fin Surface		Over-all Dimensions of Casing		Sq. Ft. of Heating Surface
		Across Tubes in In.	Along Tubes in In.	Across Tubes in In.	Along Tubes	
2	2.74	16⅝	23½	20 9/16	2'8½"	27
2½	3.43	16⅝	29½	20 9/16	3'2½"	33
3	4.14	16⅝	35½	20 9/16	3'8½"	39
3½	4.82	16⅝	41½	20 9/16	4'2½"	45
4	5.54	16⅝	47½	20 9/16	4'8½"	51
4½	6.24	16⅝	53½	20 9/16	5'2½"	57
5	6.91	16⅝	59½	20 9/16	5'8½"	64
5½	7.64	16⅝	65½	20 9/16	6'2½"	70
6	8.34	16⅝	71½	20 9/16	6'8½"	77
7	9.74	16⅝	83½	20 9/16	7'8½"	89

18-Tube Face—29-Inch Casing

Length of Tubes Feet	Net Face Area* Sq. Ft.	Inside Dimensions Over Fin Surface		Over-all Dimensions of Casing		Sq. Ft. of Heating Surface
		Across Tubes in In.	Along Tubes in In.	Across Tubes in In.	Along Tubes	
2	4.09	25 1/16	23½	29	2'8½"	40
2½	5.14	25 1/16	29½	29	3'2½"	49
3	6.18	25 1/16	35½	29	3'8½"	58
3½	7.22	25 1/16	41½	29	4'2½"	68
4	8.28	25 1/16	47½	29	4'8½"	77
4½	9.31	25 1/16	53½	29	5'2½"	86
5	10.36	25 1/16	59½	29	5'8½"	96
5½	11.40	25 1/16	65½	29	6'2½"	105
6	12.45	25 1/16	71½	29	6'8½"	115

*Net Face Area means only that area facing the tubes and does not include the headers or casings.

Note.—Depth of casings of all standard units is 10 inches in the direction of air flow. Depth of casings of all narrow units is 5 inches in the direction of air flow.

Table 97. Final Temperatures and Condensations
5 lbs. Steam—227°F. Temperature

Flexitube Aerofin-81

Temp. Entering Air	Units Deep	Velocity of Air through **Net Face Area***—in feet per minute—Measured at 70° F. and 29.92" Barometer							
		300-Ft. Face Velocity		400-Ft. Face Velocity		500-Ft. Face Velocity		600-Ft. Face Velocity	
		Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H. S.
−20°	1	62.7	1.68	54.0	2.01	47.5	2.29	42.6	2.55
0°	1	76.0	1.55	68.0	1.84	62.0	2.10	57.5	2.34
+30°	1	96.0	1.34	89.0	1.60	83.8	1.83	79.9	2.03
+40°	1	102.6	1.27	96.0	1.52	91.1	1.73	87.4	1.93
+60°	1	116.0	1.14	110.1	1.36	105.6	1.55	102.3	1.72
+70°	1	122.6	1.07	117.1	1.28	112.9	1.46	109.8	1.62

*Net Face Area means only that area facing the tubes and does not include the headers or

Table 98. Flexitube Aerofin—81 and 82
Physical Data
12-Tube Face—20 9/16-Inch Casing

Length of Tubes in Ft.	Net Face Area,* Sq. Ft.		Inside Dimensions Over Fin Surface			Over-all Dimensions of Casing		Sq. Ft. of Heating Surface	
	No. 81	No. 82	Across Tubes in Inches		Along Tubes in In.	Across Tubes in In.	Along Tubes	No. 81	No. 82
			No. 81	No. 82					
2	2.74	2.83	16⅝	17 9/16	23½	20 9/16	2'8½"	47	94
2½	3.43	3.56	16⅝	17 9/16	29½	20 9/16	3'2½"	58	116
3	4.14	4.28	16⅝	17 9/16	35½	20 9/16	3'8½"	69	138
3½	4.82	5.00	16⅝	17 9/16	41½	20 9/16	4'2½"	80	160
4	5.54	5.72	16⅝	17 9/16	47½	20 9/16	4'8½"	91	182
4½	6.24	6.44	16⅝	17 9/16	53½	20 9/16	5'2½"	102	204
5	6.91	7.16	16⅝	17 9/16	59½	20 9/16	5'8½"	114	228
5½	7.64	7.88	16⅝	17 9/16	65½	20 9/16	6'2½"	125	250
6	8.34	8.6	16⅝	17 9/16	71½	20 9/16	6'8½"	136	272

18-Tube Face—29-Inch Casing

Length of Tubes Feet	Net Face Area,* Sq. Ft.		Inside Dimensions Over Fin Surface			Over-all Dimensions of Casing		Sq. Ft. of Heating Surface	
	No. 81	No. 82	Across Tubes in Inches		Along Tubes in In.	Across Tubes in In.	Along Tubes	No. 81	No. 82
			No. 81	No. 82					
2	4.09	4.24	25 1/16	26	23½	29		70	140
2½	5.14	5.33	25 1/16	26	29½	29		87	173
3	6.18	6.41	25 1/16	26	35½	29		103	206
3½	7.22	7.50	25 1/16	26	41½	29		120	240
4	8.27	8.58	25 1/16	26	47½	29		137	273
4½	9.31	9.66	25 1/16	26	53½	29		153	306
5	10.36	10.73	25 1/16	26	59½	29		170	340
5½	11.40	11.82	25 1/16	26	65½	29		187	373
	12.45	12.90	25 1/16	26	71½	29		203	406

*Net Face Area means only that area facing the tubes and does not include the headers or casings.

Note.—Depth of casings of all standard units is 10 inches in the direction of air flow. Depth of casings of all narrow units is 5 inches in the direction of air flow.

Table 99. Constants for Obtaining Temperature Rise at Various Steam Pressures and Entering Air Temperatures not Shown in Tables

Note: To obtain Final Temperature for Steam Pressures or Entering Air Temperatures other than those shown in Tables, multiply the ratings for Zero Entering Air Temperatures and 5 Pounds Steam Pressure as shown in the various Tables, by Conversion Factors given in this Table, and add this Temperature Rise to Temperature of Entering Air.

Enter. Air Temp. °F.	Steam Pressure in Pounds per Square Inch (Gauge)															
	0	2	5	10	15	20	30	40	50	60	80	100	125	150	175	200
—20	1.021	1.050	1.088	1.142	1.187	1.227	1.295	1.350	1.399	1.441	1.514	1.575	1.641	1.699	1.750	1.795
—10	.977	1.006	1.044	1.098	1.143	1.183	1.250	1.306	1.355	1.397	1.470	1.531	1.597	1.655	1.705	1.751
0	.933	.962	1.000	1.054	1.100	1.139	1.206	1.262	1.310	1.353	1.426	1.487	1.554	1.611	1.662	1.707
10	.889	.918	.956	1.010	1.055	1.095	1.163	1.219	1.266	1.309	1.382	1.443	1.510	1.566	1.618	1.664
20	.845	.874	.912	.966	1.012	1.051	1.119	1.174	1.223	1.265	1.338	1.399	1.465	1.523	1.574	1.620
30	.801	.830	.868	.922	.968	1.007	1.075	1.130	1.179	1.221	1.294	1.355	1.421	1.479	1.530	1.575
40	.757	.786	.824	.877	.923	.963	1.030	1.086	1.134	1.177	1.250	1.311	1.378	1.435	1.486	1.531
45	.735	.764	.802	.856	.901	.941	1.009	1.064	1.113	1.155	1.228	1.289	1.355	1.413	1.464	1.510
50	.713	.742	.780	.834	.879	.919	.986	1.042	1.091	1.133	1.201	1.267	1.334	1.390	1.442	1.487
55	.691	.720	.758	.812	.857	.897	.965	1.020	1.069	1.111	1.184	1.245	1.311	1.369	1.420	1.465
60	.669	.698	.736	.790	.835	.875	.943	.998	1.046	1.089	1.161	1.224	1.290	1.346	1.398	1.444
65	.647	.676	.714	.768	.813	.853	.921	.976	1.025	1.067	1.141	1.201	1.267	1.325	1.376	1.421
70	.625	.654	.692	.746	.791	.831	.899	.954	1.003	1.045	1.119	1.179	1.245	1.302	1.354	1.400
75	.603	.632	.670	.724	.769	.809	.877	.932	.981	1.023	1.097	1.157	1.224	1.280	1.332	1.377
80	.581	.610	.648	702	.747	.787	.855	.910	.959	1.001	1.075	1.135	1.201	1.259	1.310	1.355

Table 100. Final Temperatures and Condensations
5 lbs. Steam—227°F. Temperature
Flexitube Aerofin-72

Temp. Entering Air	Units Deep	Velocity of Air through **Net Face Area*** —in feet per minute—Measured at 70°F. and 29.92″ Barometer											
		300-Ft. Face Velocity		400-Ft. Face Velocity		500-Ft. Face Velocity		600-Ft. Face Velocity		700-Ft. Face Velocity		800-Ft. Face Velocity	
		Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H.S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H.S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H.S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H.S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H.S.	Final Temp. Air	Cond. Lbs. per Hr. per Sq. Ft. H.S.
−20°	1	98.5	1.47	89.5	1.81	81.5	2.10	76.0	2.38	71.8	2.66	66.3	2.85
0°	1	108.9	1.35	100.6	1.66	93.2	1.93	88.1	2.19	84.4	2.44	79.3	2.62
+30°	1	124.6	1.17	117.4	1.45	111.0	1.67	106.5	1.90	103.3	2.12	98.8	2.27
+40°	1	129.7	1.11	122.9	1.37	116.8	1.59	111.6	1.78	109.6	2.01	105.3	2.16
+60°	1	140.1	.99	134.1	1.23	128.6	1.42	124.9	1.61	121.1	1.77	118.4	1.93
+70°	1	145.4	.94	139.6	1.15	134.5	1.33	131.0	1.51	128.4	1.69	124.9	1.82

*Net Face Area means only that area facing the tubes and does not include the headers or casings.

Table 101. Flexitube Aerofin—71 and 72
Physical Data
12-Tube Face—20⁹⁄₁₆-Inch Casing

Length of Tubes Feet	Net Face Area,* Sq. Ft.		Inside Dimensions Over Fin Surface			Over-all Dimensions of Casing		Sq. Ft. of Heating Surface	
	No. 71	No. 72	Across Tubes in Inches		Along Tubes in In.	Across Tubes in In.	Along Tubes	No. 71	No. 72
			No. 71	No. 72					
2	2.74	2.83	16⅝	17⁹⁄₁₆	23½	20⁹⁄₁₆	2′8½″	38	76
2½	3.43	3.56	16⅝	17⁹⁄₁₆	29½	20⁹⁄₁₆	3′2½″	47	94
3	4.14	4.28	16⅝	17⁹⁄₁₆	35½	20⁹⁄₁₆	3′8½″	56	112
3½	4.82	5.00	16⅝	17⁹⁄₁₆	41½	20⁹⁄₁₆	4′2½″	65	130
4	5.54	5.72	16⅝	17⁹⁄₁₆	47½	20⁹⁄₁₆	4′8½″	74	148
4½	6.24	6.44	16⅝	17⁹⁄₁₆	53½	20⁹⁄₁₆	5′2½″	83	166
5	6.91	7.16	16⅝	17⁹⁄₁₆	59½	20⁹⁄₁₆	5′8½″	92	184
5½	7.64	7.88	16⅝	17⁹⁄₁₆	65½	20⁹⁄₁₆	6′2½″	101	202
6	8.34	8.6	16⅝	17⁹⁄₁₆	71½	20⁹⁄₁₆	6′8½″	110	220

18-Tube Face—29-Inch Casing

Length of Tubes Feet	Net Face Area,* Sq. Ft.		Inside Dimensions Over Fin Surface			Over-all Dimensions of Casing		Sq. Ft. of Heating Surface	
	No. 71	No. 72	Across Tubes in Inches		Along Tubes in In.	Across Tubes in In.	Along Tubes	No. 71	No. 72
			No. 71	No. 72					
2	4.09	4.24	25¹⁄₁₆	26	23½	29	2′8½″	57	113
2½	5.14	5.33	25¹⁄₁₆	26	29½	29	3′2½″	70	140
3	6.18	6.41	25¹⁄₁₆	26	35½	29	3′8½″	84	167
3½	7.22	7.50	25¹⁄₁₆	26	41½	29	4′2½″	97	194
4	8.27	8.58	25¹⁄₁₆	26	47½	29	4′8½″	111	221
4½	9.31	9.66	25¹⁄₁₆	26	53½	29	5′2½″	124	248
5	10.36	10.73	25¹⁄₁₆	26	59½	29	5′8½″	138	275
5½	11.40	11.82	25¹⁄₁₆	26	65½	29	6′2½″	151	302
6	12.45	12.90	25¹⁄₁₆	26	71½	29	6′8½″	165	329

*Net Face Area means only that area facing the tubes and does not include the headers or casings.

Note. Depth of casings of all standard units is 10 inches in the direction of air flow. Depth of casings of all narrow units is 5 inches in the direction of air flow.

Items Necessary for Final Selection of Heating Coils.* When selecting a heating unit for any particular case the choice should be based on the following items:

(1) Final temperature desired.

(2) Loss in pressure (or friction) of air passing over heating coil. The manufacturers' tables give such information. (See Table 95.)

(3) Air velocity over heating unit. This item is generally given in manufacturers' tables as being measured at 70°F. For other temperatures the manufacturers' catalogues should be consulted.

(4) Free area or face area of heating unit.

(5) Ratio of heating surface to net free (or face) area.

(6) Air volume required.

(7) Number of rows of pipes, tubes, or sections. Shown in manufacturers' catalogues.

(8) Amount of heating surface.

(9) Steam pressure drop through the heating unit. Shown in manufacturers' catalogues.

(10) Weight of heating unit. This must be considered because of loading specifications on various floors.

Piping Diagram. Fig. 219 shows a typical and recommended pipe diagram for a battery of Aerofin Heating Coils three units wide and one row deep and where the tubes are vertical. Such a diagram is applicable for steam pressures below 20 pounds and can be used in Open Circuit Gravity, Vented Return, or Vacuum Systems. Fig. 219 is shown in isometric to help the reader better visualize the arrangement of pipes. Manufacturers' catalogues give complete data for all combinations of heating coils.

Water Cooling Coils.† The use of water through coils for cooling and air conditioning is recognized by engineers as an economical method of providing comfort and process conditioning in many localities. The greatest number of cooling installations made use cold water, circulating through the coils for direct cooling application. This is true of installations where water is not available from wells or mains at a low enough temperature as well as where water is available during the cooling season at 60° or lower. Particularly is it true where the system is to be used for both heating and cooling because on such installations the use of a commercial refrigerant direct to the coils requires the use of separate coils for winter heating. Usually on installations of this type the compressor or steam jet equipment is

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*Courtesy of A.S.H.V.E. Guide, 1935, Chapter 22.
†Data Courtesy of the Trane Co., La Crosse, Wis.

used to cool the water to a usable temperature and the water is then piped to the units. On railroad car equipment ice has been very popular as a cooling medium. Here again chilled water is piped to the cooling units to provide air conditioning.

The cost of installing and operating air-conditioning systems in many communities can be reduced considerably by using cooling
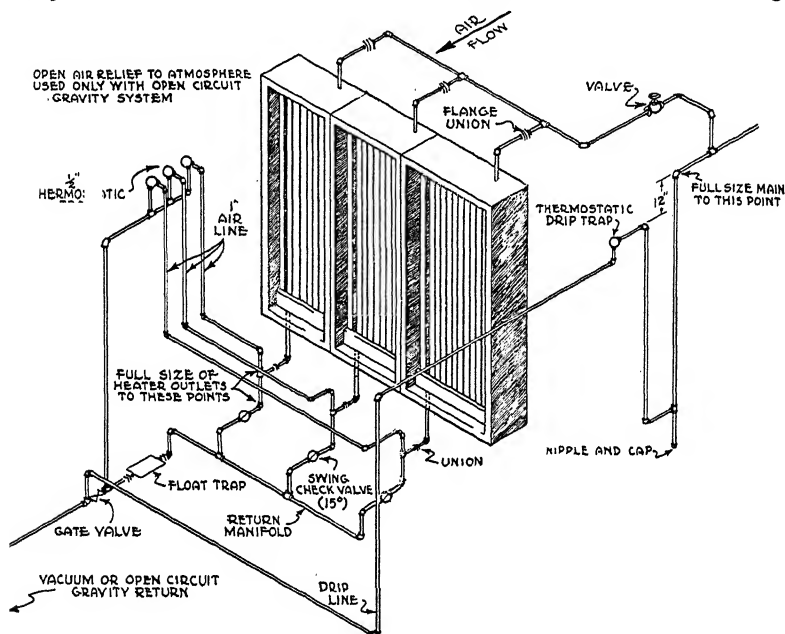
Fig. 219. Piping Diagram for Battery of Aerofin Heating Coils
Three Units Wide and One Unit Deep

coils with water as the cooling medium instead of commercial refrigerants.

Coils are installed in residential, commercial, and industrial applications of air conditioning where the sole cooling medium is water. Very fine results are secured if the water is available during the cooling season at temperatures of 60° or lower and it is possible with an installation of this type to reduce the leaving air temperature to within a few degrees of the entering water temperature.

On other installations a very economical and satisfactory installation is secured by using the water through a cooling coil to pre-cool the air before it enters the direct expansion coil.

Water coils are available in two types, type "E" for duct work connection on central systems and type "C" for commercial cooling applications in cabinets, and similar installations. Fig. 220 shows a cutaway view of a typical water cooling coil. It shows also the header and casing construction of Type E coils, including the guide flange assembly. This design permits the heater core to expand and contract independently of the casing, and without strain on the tubes. The $\frac{1}{8}$-inch steel guide angle (1) is rigidly bolted to the cast-iron header (4) and slides between the spacer angles (2) and the web of
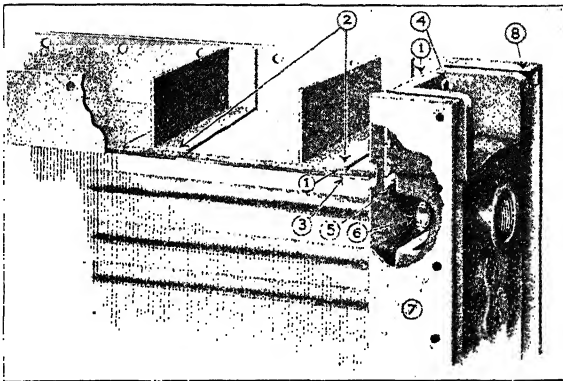
Fig. 220. Trane Type E Water Cooling Coil

the side channel (3). The header is independent of the end channel (7) and the oversize holes (6) permit the tubes to move freely. Accordingly, the heater core, consisting of fins, tubes, and headers, floats in the casing. Swing joints are recommended in all piping connections to prevent the transmission of expansion strains from the system to the coil. However, even when this precaution is taken, there still may be some stress transmitted to the coil from the piping. The guide flange assembly protects the heater tubes by transmitting such stresses to the side channels.

This feature permits bolting the casings of Trane Type E coils rigidly to ductwork or to any mounting frame without danger of damage from expansion strains.

Comparative Cost of Cooling with Water and Direct Expansion Refrigerants. The charts shown in Figs. 221, 222, and 223 give a comparative operating cost of cooling with water and direct expansion
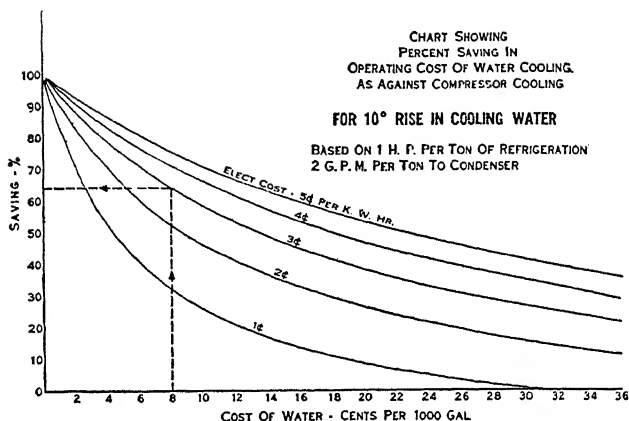
Fig. 221.   Illustrative Chart
Courtesy of The Trane Co., La Crosse, Wis.

refrigerants for varying water temperatures.   The approximate savings can be found readily by knowing the temperature rise of the water through the coils, the cost of water per 1000 gallons, and the electrical cost in the community.

Assume a problem where the water enters the coils at 54°F. and
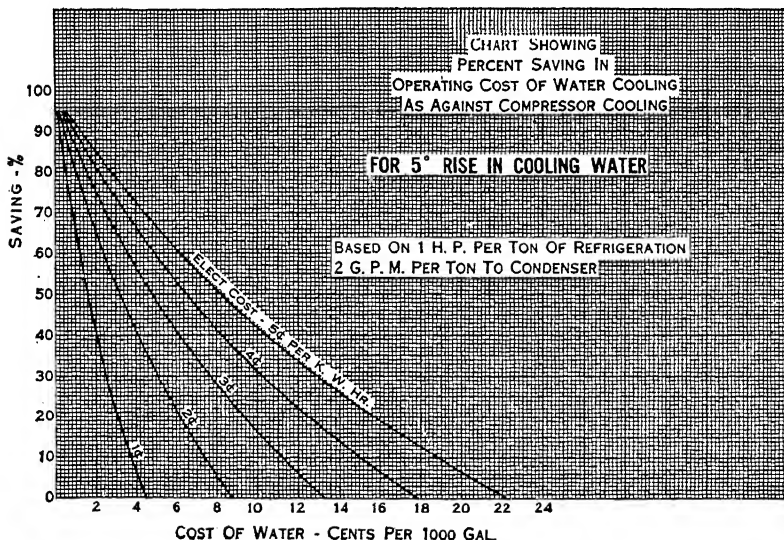
Fig. 222.   Chart for 5-Degree Rise
Courtesy of The Trane Co., La Crosse, Wis.

leaves at 64°F., which is a 10°F. temperature rise. Local water costs 8¢ per 1000 gallons and electricity 3¢ per kilowatt hour.

Referring to the illustrative chart, Fig. 221, and also to the 10°F. rise chart, Fig. 223, we find that the saving amounts to 64%.
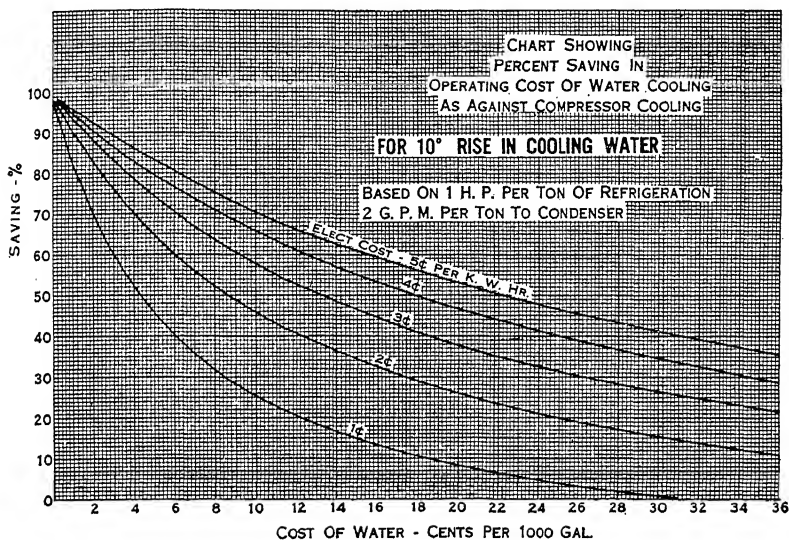
Fig. 223.   Chart for 10-Degree Rise
Courtesy of The Trane Co., La Crosse, Wis.

Assuming the same problem with only a 5°F. temperature rise we find that the saving amounts to only 27%, by referring to Fig. 222 for a 5°F. rise.

Procedure for Selecting Coils. In selecting a cooling coil, it is necessary to have the following primary data:

 (1) Dry- and wet-bulb temperature of the entering air.
 (2) Dry- and wet-bulb temperature of the leaving air.
 (3) Total c.f.m. of air to be cooled.
 (4) Water temperature and g.p.m.

With these four factors given, it is possible to calculate the size of the cooling coil required. From the above four primary factors which must be known for any problem, the following quantities required in determining the coil, can be calculated:

 (1) Total heat to be removed by the coil.
 (2) Face area of the cooling coil.
 (3) Quantity of cooling water to be circulated.
 (4) Water velocity in the tube.
 (5) Wet surface factor due to dehumidification.

(6) The mean temperature difference.

(7) Number of rows of tubes.

Example 1. The following example with primary data will illustrate the process: 10,000 c.f.m. of air is to be cooled from 82°F. dry-bulb temperature and 70°F. wet-bulb temperature to 58°F. dry-bulb temperature and 55°F. wet-bulb temperature. (Usually there is about a 3°F. difference between the dry- and wet-bulb temperature of the air leaving the coil.) Water at 50°F. in ample quantities is available for cooling.

Total Heat to Be Removed by the Coil. The total heat removed from each pound of air flowing through the coils will be the difference in the total heat content of the air at the initial and final wet-bulb temperatures.

From columns 17 and 18 of Table 90.

Total Heat of Air at 70°F. wet-bulb temperature = 33.51 B.t.u. per lb.

Total Heat of Air at 55°F. wet-bulb temperature = 23.04 B.t.u. per lb.

Heat removed per pound of air = 10.47 B.t.u.

$$\text{(Total heat to be removed by the coil)} = 4.45 \times \text{c.f.m.} \times \left\{ \begin{array}{l} \text{B.t.u. re-} \\ \text{moved per} \\ \text{lb. of air.} \end{array} \right\}$$
$$= 4.45 \times 10{,}000 \times 10.47$$
$$= 467{,}000 \text{ B.t.u. per hour.}$$

Selecting the Coil Face Area. The face area of the coil can be determined from the following formula:

$$\text{Face area} = \frac{\text{c.f.m.}}{\text{velocity}}$$

In this problem we are given the quantity of air to be supplied as 10,000 c.f.m. For average installations use a face velocity from 400 to 600 feet per minute. In order to keep the air friction loss through the coils from becoming too large, the lower velocities should be used with the larger number of rows or tubes. Assume for this problem a velocity of 500 feet per minute.

$$\text{Face area} = \frac{10{,}000 \text{ c.f.m.}}{500 \text{ feet per minute velocity}}$$
$$= 20 \text{ square feet}$$

Referring to Table 102, we find that two 24″×60″ coils will have a face area of 20 square feet. Therefore, two coils each having a 24″ header and tubes 60″ long will satisfy the conditions of this problem. All that remains now in determining the complete coil is to find the required number of rows.

Quantity of Cooling Water to Be Circulated. The quantity of cooling water to be circulated depends entirely upon the allowable temperature rise of the water. The greater the allowable temperature rise of the water, the smaller the quantity of water which must be circulated. Table 103 shows this clearly. However, with counterflow the allowable temperature rise is strictly limited by the difference between the initial temperature of the air and the final temperature of the water. This difference should be at least 15°F. and should be 20°F. preferably. The greater the difference the less surface will be required. There are instances, of course, in which it is necessary to use a smaller difference than 15°F., in which case it is necessary to install a larger number of rows of tubes or increase the quantity of water to do the cooling job.

The difference between the final temperature of the air and the initial temperature of the water should not, as a rule, be less than 5°F., and should be 7°F. or 8°F. preferably, if it is desired to use as little cooling surface as possible. This does not mean that we cannot cool the air down to within 3°F. of the initial water temperature if the occasion arises. It is only a question of providing a sufficient number of rows of tubes, or enough water.

For this problem we have 50°F. water available. Assume also that we are going to allow the water to heat up to a temperature within about 20°F. of the entering air. Since the dry-bulb temperature of the air entering the coils is 82°F., we will provide such a quantity of water that 467,000 B.t.u.'s per hour (the total heat to be removed by the coil) can heat the circulating water to a final temperature of 62°F., which is 20°F. below 82°F., the initial air temperature. The rise in water temperature will be

$$62°F. - 50°F. = 12°F.$$

From Table 103 we find that for a 12°F. rise in water temperature we must circulate 0.167 g.p.m. for each 1000 B.t.u.'s per hour of total heat which must be removed by the coil. We found that the total heat to be removed by the coil was 467,000 B.t.u.'s per hour. The total g.p.m. to be circulated is, therefore

$$\text{g.p.m.} = 0.167 \times \frac{467,000}{1000} = 78$$

The quantity of water can be found by the following formula:

$$\text{g.p.m.} = \frac{\text{total heat}}{500 \times \text{temperature rise}}$$

Therefore

$$\text{g.p.m.} = \frac{467,000}{500 \times 12} = 78$$

If the water quantity available for cooling is limited, then the coil selected must have a sufficient number of rows of tubes to allow this small quantity of water to rise through a large enough temperature range to absorb the heat to be removed from the air. But, the final temperature of the water must in any case be less than the temperature of the entering air.

Water Velocity in Tubes. The water velocity in the coils must be calculated to find the heat transfer factor and friction head lost. The water velocity in the coils is found from Fig. 224. In using this chart remember that the supply of 78 g.p.m. is split between two 24-inch headers which means that we have a flow of 39 g.p.m. through each header. We might have selected a 30×96-inch coil from Table 101 which would also have given a face area of 20 square feet. In that case the entire 78 g.p.m. would flow through one 30-inch header. However, we will continue our problem on the basis of two 24-inch headers with a flow of 39 g.p.m. through each header. Referring to Fig. 224 we see that the water velocity for 39 g.p.m. through a 24-inch header is 2.4 feet per second.

Wet Surface Factor Due to Dehumidification. To find the wet surface factor, we need the following quantities:

(1) The dry-bulb temperature of the air entering the coils.

(2) The dew-point temperature of the air entering the coils.

(3) The temperature of the water entering the coil.

In this problem, the dry-bulb temperature of the air entering the cooling coils is 82°F. and the wet-bulb temperature is 70°F. Referring to the Psychrometric Chart (found in the back of the book) we find that the dew point of the air entering the coil is 64°F. We found that the temperature of the water entering
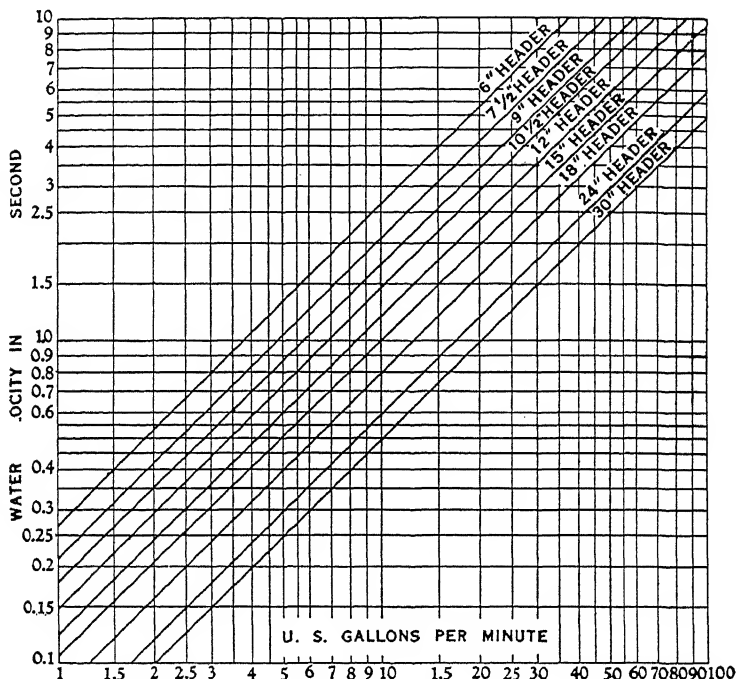
Fig. 224. Chart Showing Water Velocities
Courtesy of The Trane Co., La Crosse, Wis.

the coils was 50°F. In order to make use of Fig. 225, we must obtain the following quantities:

Difference between entering air dry-bulb temperature and the entering water temperature.

Dry-bulb difference = 82°F. − 50°F. = 32°F.

Difference between entering air dew point and the entering water temperature:

Dew point difference = 64°F. − 50°F. = 14°F.

Referring to Fig. 225, we find that for 32°F. and 14°F., the wet surface factor is 1.2.

Mean Temperature Difference. Inasmuch as the water should always be introduced counterflow to the air, the mean temperature difference (usually abbreviated m.t.d.) is determined as follows:

(1) Subtract the temperature of the leaving water from the entering air.

(2) Subtract the temperature of the entering water from the leaving air.
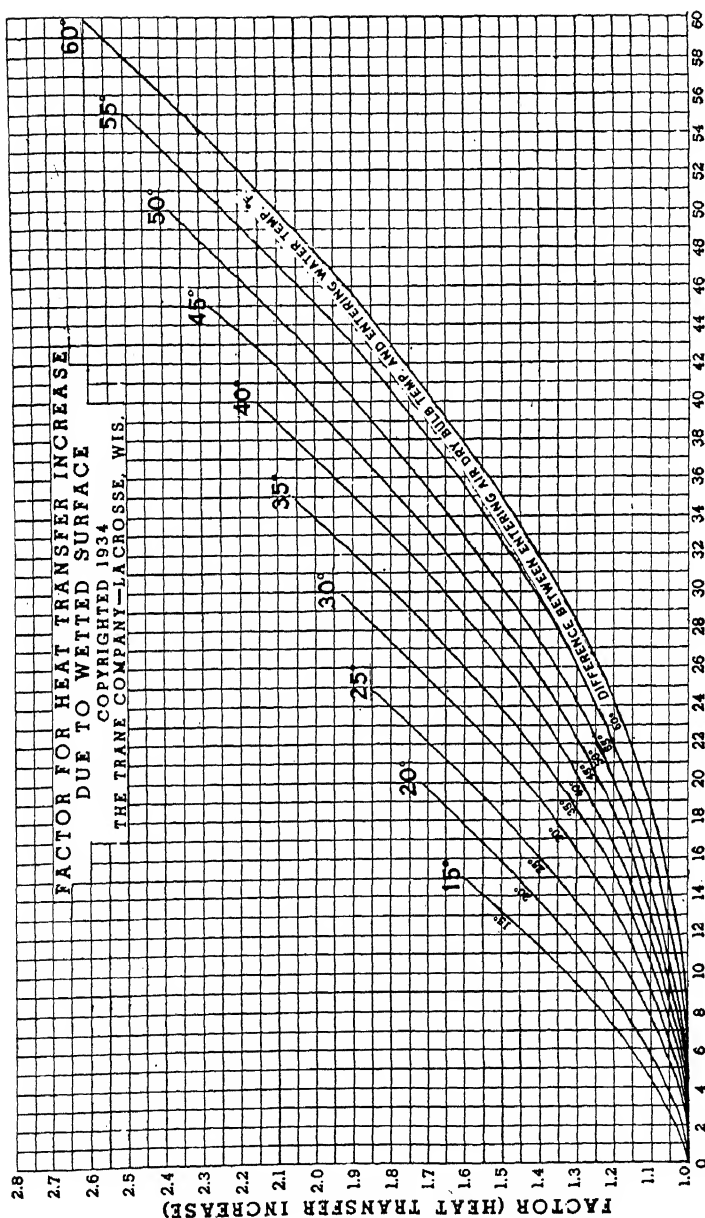
Fig. 225. Chart Showing Factor for Heat Transfer Increase

$$82°F. \xrightarrow{\text{Air}} \; 58°F.$$
$$62°F. \xleftarrow{} \; 50°F.$$
$$\text{Water}$$
$$20°F. \quad \text{Difference} \quad 8°F. \text{ Difference}$$

From the Table 91 we find, for an 8°F. and 20°F. difference that the m.t.d. is 13.2°F.

Number of Rows of Tubes Required. The number of rows of tubes required is found from the following formula:

$$\text{Rows of Tubes} = \frac{\text{total heat to be removed by coil}}{\left(\begin{array}{l} \text{face area} \times \text{m.t.d.} \times \text{heat transfer} \\ \text{capacity} \times \text{wetted surface factor} \end{array} \right)}$$

From Table 104 for an air velocity of 500 feet per minute and a water velocity of 2.4 feet per second, the heat transfer capacity is 186 B.t.u.

$$\text{Rows of Tubes} = \frac{467,000}{20 \times 13.2 \times 186 \times 1.2} = 7.9$$
$$\text{or 8 rows of tubes}$$

The face dimensions of our cooling surface will be 48″ high by 60″ long. Each coil will be 8 rows of tubes deep. The cooling coils will consist of eight 24×60-inch coils, each 2 rows deep. The coils will be arranged 2 high by 4 deep. The over-all dimension of the coils can be found in Table 105.

In cases where the above method gives an odd number of rows of tubes, such as 5 or 7, it is better to use the next highest number of even rows, such as 6 or 8, respectively. Inasmuch as there are 2 rows of tubes to a header, a 7-row coil would require 4 headers in any event. The additional cost of 8 rows of tubes in the fourth header is very small.

In addition there is the question of increased water friction in the last coil if only a single row of tubes is used in this header. For example, suppose 78 g.p.m. of water are used in a 7-row coil. The first 3 headers, each having 2 rows of tubes would split the 78 g.p.m. between each row of tubes; that is, 39 g.p.m. would flow through each row of tubes. But the last coil having only one row of tubes would pass the entire 78 g.p.m. through the single row of tubes. As a result, the water friction head lost in the fourth coil alone would be greater than the total head lost in the entire first 3 coils having 6 rows of tubes.

Also, where the formula shows that an odd number of rows of tubes are required, it is possible to drop back to the next lowest even number of rows of tubes by either increasing the quantity of circulating water or decreasing the air velocity. Decreasing the air velocity means that you will be increasing the face area of the coil. Thus if the formula shows that you require 7 rows of tubes, you can remove the same amount of heat with 6 rows by either increasing the quantity of circulating water or increasing the face area of the coil.

Air Friction through Coils. The friction loss of air flowing at 500 feet per minute velocity through 8 rows of tubes is shown by Table 106 to be 0.327 inches.

The friction loss through the coil will be increased due to the condensation of moisture on the surface. Table 106 gives the increase. We found the difference between the dew point of the entering air and the refrigerant to be 14°F. Referring

to Table 107, we find for a 14° difference that the friction factor is 1.14. Therefore our actual friction drop will be:

$$1.14 \times .327 = .373 \text{ inches of water}$$

Water Friction through Coils. Since each coil has tubes 60 inches (or 5 feet) long, and there are four headers (one header for each 2 rows of tubes) the total length of tubing through which the water must flow is:

$$4 \times 5 = 20 \text{ feet}$$

Referring to Fig. 226 we see that for a water velocity of 2.4 feet per second the friction loss is 2.7 feet for each 10-foot length of tubing. Therefore the total water friction drop through the coil will be:

$$2.7 \times \frac{20}{10} = 5.4 \text{ feet of water head.}$$

Interconnecting manifolds between the sections will probably add about 30% to the above figure which will give the actual head lost:

$$5.4 \times 1.3 = 7 \text{ feet of water head.}$$

Example 2. The selection of the correct amount of cooling surface depends only upon the four factors given in Example 1. However, the correct determination of these four factors is in itself a problem, which unless correctly solved will result in an unsatisfactory installation. For this reason a complete solution of an air-conditioning calculation, illustrating how the various related factors are determined, is given here.

Before selecting a cooling coil, it is necessary to know or to determine the following quantities:

(1) Sensible and latent heat gains of the conditioned space.

(2) Dry- and wet-bulb temperature of the air entering the coils.

(3) Dry- and wet-bulb temperature of the air leaving the coils.

(4) c.f.m. of air to be cooled and delivered to the conditioned space.

(5) Total heat to be removed by the coil.

(6) g.p.m. of cooling water to be circulated.

After the above quantities have been determined, the selection of the proper size coils is a fairly simple matter. For the sake of completeness and clarity, the method of determining the above quantities is presented here as well as the method of selection of the coil.

80°F. dry-bulb temperature and 50% relative humidity is so rapidly becoming the assumed inside air condition universally used for design purposes that Tables 108 and 109 based on this inside condition have been prepared.

With these tables the following items can be determined directly:

(1) Dry- and wet-bulb temperature of the air entering the coils for any percentage of outside air.

(2) Required dry- and wet-bulb temperature of the air leaving the cooling coils.

(3) c.f.m. of air to be supplied to the conditioned space.

(4) Coil face area required.

Sensible and Latent Heat to Be Removed. The sensible and latent heat to be removed is found from a calculation of the heat gains of the room. Assume in this

case that the sensible and latent heat gains of the room have been figured and amount to:

Sensible heat gain = 165,000 B.t.u.'s per hour
Latent heat gain = 30,000 B.t.u.'s per hour
Total heat gain   = 195,000 B.t.u.'s per hour

The above figures do not include the sensible and latent heat to be removed in cooling the outside air, which is introduced for ventilation purposes. How the heat of the ventilation air is taken care of will be shown in the next step.

Dry- and Wet-Bulb Temperature of the Air Entering the Coil. Assume that the following room conditions are to be maintained:

80°F. room dry-bulb temperature
67°F. room wet-bulb temperature
60°F. room dew-point temperature
50% room relative humidity

If no outside air were supplied to the continued room for ventilation purposes, all the air entering the coils would be withdrawn from the room, in which case the dry- and wet-bulb temperature of the air entering the coils would be the same as in the conditioned room, that is 80°F. and 67°F. respectively.

If outside air is to be supplied to the conditioned room for ventilation purposes, it is best to mix the outside air with return air from the conditioned room ahead of the cooling coils and then cool the mixture of the outside air and return air. Since the outside air is warmer than the room air, it is evident that a mixture of the two will have a higher dry-bulb temperature than the return air alone. To determine the resultant dry- and wet-bulb temperature, it is only necessary to know what percentage of the total air drawn through the coil is taken from the outside.

Assume that the requirements of the job call for an outside air supply equal to 20% of the total air delivered to the conditioned room. The outside air has a 95°F. dry-bulb temperature and 75°F. wet-bulb temperature.

Twenty per cent of the total air supplied to the conditioned room was to be outside air. Refer to Table 108. In Column 2 we find the dry-bulb temperature of the mixture entering the cooling coils is 83°F. In Column 6 we find the wet-bulb temperature of the mixture entering the cooling coils is 68.7°F.

Dry- and Wet-Bulb Temperature of the Air Leaving the Coil. The dry- and wet-bulb temperature of the air leaving the coils is of course determined by finding the dry- and wet-bulb temperature at which it is necessary to introduce the air supply into the conditioned room to absorb the sensible and latent heat gains of the room. We have given the following heat gains in the conditioned room:

Sensible heat gain = 165,000 B.t.u.'s per hour
Latent heat gain = 30,000 B.t.u.'s per hour
Total heat gain   = 195,000 B.t.u.'s per hour

$$\text{Sensible heat percentage} = \frac{\text{Sensible heat gain}}{\text{total heat gain}}$$

$$= \frac{165,000}{195,000}$$

$$= 84.8\%$$

Refer to Table 109. In Column 9 for a sensible heat percentage of 85% we find the required wet-bulb temperatures of air supply for various dry-bulb temperatures. If we supply saturated air the temperature would have to be 57.8°F. However, there is usually about a 3 degree difference between the dry- and wet-bulb temperature of the air leaving the coil. Therefore we will choose a dry-bulb temperature such that the required wet-bulb temperature will be about 3 degrees lower. Referring to Table 108 again, we see that an air supply having a dry-bulb temperature of 64°F. will require a wet-bulb temperature of 60.5°F. Therefore, the air leaving the cooling coils must have a dry-bulb temperature of 64°F. and a wet-bulb temperature of 60.5°F.

C.f.m. of Air to Be Supplied. Referring to Column 2 of Table 108, we see that for an air supply having a dry-bulb temperature of 64°F., we must supply 56.55 c.f.m. for each 1000 B.t.u.'s of sensible heat gain per hour. Since the sensible heat gain of this example is 165,000 B.t.u.'s per hour, we find the total c.f.m. as follows:

$$\text{Total c.f.m. to be supplied} = 56.55 \times \frac{165,000}{1000}$$

$$= 9,350$$

Total Heat to Be Removed by the Coil. If no outside air were introduced into the conditioned room for ventilation purposes, the total load on the coil would only be equal to the total heat gains of the room, or in this case 195,000 B.t.u.'s per hour. But inasmuch as outside air is mixed with the return air ahead of the coil, we have seen that the wet-bulb temperature of the mixture entering the coils is increased to 68.7°F. The wet-bulb temperature of air leaving the coil was found to be 60.5°F.

The total heat removed per pound of air by the cooling coil will be the difference in the total heat content of the air at the initial and final wet-bulb temperatures of the air. From Columns 11 and 9 of Table 90.

Total heat of air at 68.7°   = 32.47 B.t.u.'s per lb.
Total heat of air at 60.5°   = 26.51
Heat removed per pound of air = 5.96 B.t.u.'s

We found that an air supply of 9350 c.f.m. was required for the conditioned

$$\begin{pmatrix}\text{Total heat to be} \\ \text{removed by coil}\end{pmatrix} = 4.45 \times \text{c.f.m.} \times \begin{pmatrix}\text{B.t.u.'s removed} \\ \text{per pound of air}\end{pmatrix}$$

$$= 4.45 \times 9350 \times 5.96$$

$$= 248,000 \text{ B.t.u.'s per hour to be removed by the coil}$$

The increase from 195,000 to 248,000, that is, 53,000 B.t.u.'s, represents the heat removed in cooling the outside air.

G.p.m. of Cooling Water to Be Circulated. The quantity of cooling water to be circulated depends entirely upon the allowable temperature rise of the water. The greater the allowable temperature rise of water, the smaller the quantity of water which must be circulated. Table 103 shows this clearly. However, the allowable temperature rise is strictly limited by the difference between the initial temperature of the air and the final temperature of the water. In general this

difference should be roughly at least 15°F. and preferably 20°F. The greater the
difference the less surface you will require. There are instances of course in which
it is necessary to use a smaller difference than 15°F., in which case it will be
necessary to install a larger number of rows of tubes or increase the quantity
of water to do the cooling job.

Incidentally the difference between the final temperature of the air and the
initial temperature of the water should never be less than 5 degrees, and should
preferably be 7 or 8 degrees, in order to use as little surface as possible. This does
not mean that we cannot cool the air down to within 3 degrees of the initial water
temperature if the occasion arises. It is only a question of providing a sufficient
number of rows of tubes or enough water.

For this example assume that we have 56°F. water available. Assume that
we are going to allow the water to heat up to a temperature within about 20 de-
grees of the entering air. We found that the dry-bulb temperature of the air
entering the coils is 83°F. Therefore, we will provide such a quantity of water
that 248,000 B.t.u.'s per hour (the total heat to be removed by the coil) can
heat the circulating water to a final temperature of 64°F. which is 19 degrees
below 83°F. The rise in water temperature will be:

$$64°F. - 56°F. = 8°F.$$

From Table 103, we find that for an 8 degree rise in water temperature we
must circulate 0.25 g.p.m. for each 1000 B.t.u.'s per hour of total heat to be re-
moved by the coil. We found that the total heat to be removed by the coil was
248,000 B.t.u.'s per hour. The total g.p.m. to be circulated is, therefore

$$\text{g.p.m.} = 0.25 \times \frac{248,000}{1000} = 62$$

Selecting Cooling Coil. To select the cooling coil, we must determine the
following factors just as in Example 1: (1) the face area of the coil; (2) the
water velocity in the tubes; (3) the wet surface factor; and (4) the mean tem-
perature difference.

Face Area of Coil. The face area of the coil can be determined from the
following formula:

$$\text{Face area} = \frac{\text{c.f.m.}}{\text{Velocity}}$$

We found that the quantity of air to be supplied is 9,350 c.f.m. For average
installations use a face velocity from 400 to 600 feet per minute. To keep the air
friction loss through the coils from becoming too large, the lower velocities should
be used with the larger number of rows of tubes. Assume for this problem that
we will use a velocity of 500 feet per minute.

$$\text{Face area} = \frac{9,350}{500} = 18.7 \text{ square feet}$$

Referring to Table 102, we find that a 30×90-inch coil has a face area of
18.75 square feet. Therefore, a coil having a 30-inch header and tubes 90″ long
will satisfy the conditions of this problem.

Water Velocity in the Tubes. It is necessary to determine the water velocity
in the tubes to find the heat transfer factor and the friction head lost. The water

velocity in the tubes is found from Fig. 224. In using this chart, remember that all of the water circulated flows through each header of the cooling coils. In this case we have a 30-inch header and Fig. 224 shows that 62 g.p.m. through a 30-inch header will give a water velocity of 3.0 feet per second through each tube.

Wet Surface Factor. To find the wet surface factor, we need the following quantities:

(1) The dry-bulb temperature of the air entering the coils.

(2) The dew-point temperature of the air entering the coils.

(3) The temperature of the water entering the coil.

We found that the dry-bulb temperature of the air entering the cooling coils was 83°F. and the wet-bulb temperature was 68.7°F.

Referring to the Psychrometric Chart (found in the back of the book), we find that the dew point of the entering air is 61°F. We found that the temperature of the water entering the coils was to be 56°F. In order to make use of Fig. 225, we must obtain the following quantities:

Difference between entering air dry-bulb temperature and the entering water temperature is:

$$\text{Dry-bulb difference} = 83°F. - 56°F. = 27°F.$$

Difference between entering air dew point and the entering water temperature is:

$$\text{Dew-point difference} = 61°F. - 56°F. = 5°F.$$

Referring to Fig. 223, we find that the wet surface factor is 1.04.

Mean Temperature Difference. Inasmuch as the water should always be introduced counterflow to the air, the mean temperature difference is determined as follows:

(1) Subtract the temperature of the leaving water from the entering air.

(2) Subtract the temperature of the entering water from the leaving air.

83°F.	Air	64°F.
64°F.	Water	56°F.
19°F. Difference		8°F. Difference

From Table 91 we find for an 8°F. and a 19°F. difference that the m.t.d. (Mean Temperature Difference) is 12.8°F.

Number of Rows of Tubes Required. The number of rows of tubes required is found from the following formula:

$$\text{Rows of Tubes} = \frac{\text{Total heat to be removed}}{\left(\begin{array}{c}\text{Face area} \times \text{m.t.d.} \times \text{heat transfer}\\ \text{factor} \times \text{wet surface factor}\end{array}\right)}$$

From Table 104, for an air velocity of 500 feet per minute and a water velocity of 3.0 feet, per second, the heat transfer capacity is 189 B.t.u.'s.

$$\text{Rows of Tubes} = \frac{248,000}{18.75 \times 12.8 \times 189 \times 1.04}$$

$$= 5.3 \text{ or } 6 \text{ rows of tubes.}$$

The coil face dimensions will be **30″** wide by **90″** long. The coil will be 6 rows of tubes deep and will have four headers, one for each two rows of tubes. The over-all dimension of the coil can be found in Table 105. We have selected 6 rows of tubes for the reasons given in Example 1.

Air Friction. The friction loss of air flowing at 500 feet per minute velocity through 6 rows of tubes is shown by Table 110 to be 0.253 inches.

The air friction loss through the coil will be increased due to the condensation of moisture on the surface. Table 107 gives the increase. In this example under "Wet Surface Factor" we found the difference between the dew point of the entering air and the entering refrigerant to be 5°F. Referring to Table 105, we find

FRICTION DROP IN FEET OF WATER FOR EACH 10 FT OF TUBE LENGTH
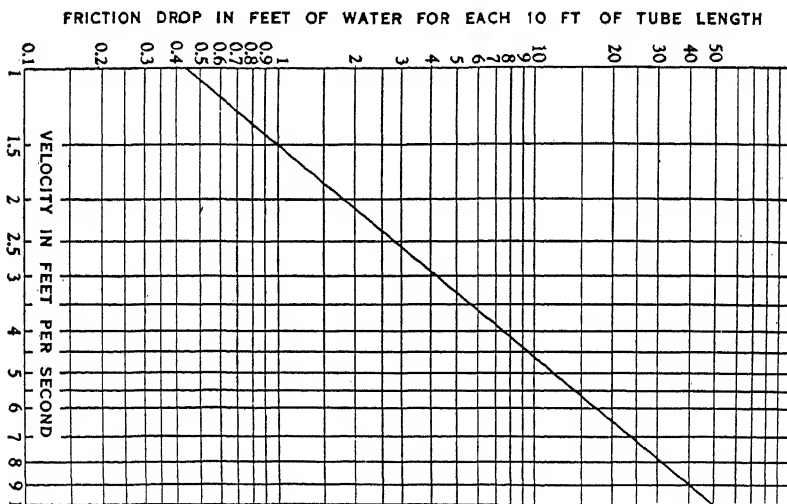
Fig. 226.   Chart Showing Water Friction
Courtesy of The Trane Co., La Crosse, Wis.

for a 5°F. difference that the friction factor is about 1.05. Therefore our actual friction drop will be:

$$1.04 \times .253 = 0.263 \text{ inches of water}$$

Water Friction. Since each coil has tubes 90″ long or 7.5 feet long, and there are three headers (one header for each two rows of tubes) the total length of tubing through which the water flows is:

$$3 \times 7.5 = 22.5 \text{ feet}$$

Referring to Fig. 226, we see for a water velocity of 3.0 feet per second that the friction head lost is 4.2 feet for each 10-foot length of tubing. Therefore the total water friction drop through the coil will be:

$$4.2 \times \frac{22.5}{10} = 9.45 \text{ feet of water head}$$

Interconnecting manifolds between the sections will probably add about 30% to the above figure which will give for the actual head lost

$$9.45 \times 1.3 = 12.3 \text{ feet}$$

Table 102. *Square Feet Net Face Area—Not Including Casing

Nominal or Order-ing Width Inches	12	18	24	30	36	42	48	54	60	66	72	78	84	90	96	102
12	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5
15	.....	1.88	2.5	3.13	3.75	4.38	5.0	5.63	6.25	6.88	7.5	8.13	8.75	9.38	10.0	10.63
18	.....	2.25	3.0	3.75	4.5	5.25	6.0	6.75	7.5	8.25	9.0	9.75	10.5	11.25	12.0	12.75
21	.....	.....	3.5	4.38	5.25	6.13	7.0	7.88	8.75	9.63	10.5	11.38	12.25	13.13	14.0	14.88
24	.....	.....	4.0	5.0	6.0	7.0	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0
30	.....	.....	.....	6.25	7.5	8.75	10.0	11.25	12.5	13.75	15.0	16.25	17.5	18.75	20.0	21.25
33	.....	.....	.....	.....	8.25	9.63	11.0	12.38	13.75	15.13	16.5	17.88	19.25	20.63	22.0	23.38

Table 103. *G.P.M. of Cooling Water to be Circulated for each 1000 B.t.u. per Hour of Total Heat Removed by the Cooling Coils

Temperature Rise of the Water	G.P.M. per 1000 B.t.u.	Temperature Rise of the Water	G.P.M. per 1000 B.t.u.
5°	.400	15°	.133
6	.333	16	.125
7	.285	17	.118
8	.250	18	.111
9	.220	19	.105
10	.200	20	.100
11	.182	21	.095
12	.167	22	.091
13	.154	23	.087
14	.143	24	.083

Table 104. *Heat Transfer Capacities of Trane Coils for Various Air and Water Velocities

Capacities are given in B.t.u. per hour per row of tubes per degree of mean temperature difference per square foot of face area—(for use only with water and brine above 32°F.).

Water Velocity Feet per Second	Air Velocity in Feet per Minute									Water Velocity Feet per Second
	200	300	400	500	600	700	800	900	1000	
.25	88	100	110	118	126	132	138	144	148	.25
.30	91	104	115	123	130	137	143	149	154	.30
.40	97	110	121	130	138	145	152	158	163	.40
.50	101	115	127	136	144	152	159	165	171	.50
.60	106	120	132	142	151	158	166	172	178	.60
.70	109	124	137	147	155	163	171	178	184	.70
.80	111	127	139	149	159	167	174	181	187	.80
.90	113	129	142	152	162	170	178	185	191	.90
1.00	116	132	145	155	165	173	181	188	195	1.00
1.10	118	134	148	158	168	177	185	192	198	1.10
1.20	120	137	150	161	171	180	188	196	202	1.20
1.30	122	139	153	164	174	183	192	199	206	1.30
1.40	124	142	156	167	177	186	195	203	210	1.40

Note: For water velocities greater than 2.6 feet per second use the heat transmission capacities given for 2.6 feet per second.

*Courtesy of The Trane Co. Complete tables can be secured from manufacturer.

Table 105. *Roughing in Dimensions

Dimension "A"=Nominal or Ordering Length

	12 In. Header		15 In. Header		18 In. Header		21 In. Header		24 In. Header	
	1 Row (Inches)	2 Row (Inches)	1 Row (Inches)	2 Row (Inches)	1 Row (Inches)	2 Row (Inches)	1 Row (Inches)	2 Row (Inches)	1 Row (Inches)	2 Row (Inches)
B	12	12	15	15	18	18	21	21	24	24
C	$16\frac{1}{2}$	$16\frac{1}{2}$	$19\frac{1}{2}$	$19\frac{1}{2}$	$22\frac{1}{2}$	$22\frac{1}{2}$	$25\frac{1}{2}$	$25\frac{1}{2}$	$28\frac{1}{2}$	$28\frac{1}{2}$
D	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$	$A+4\frac{5}{8}$
E	$5\frac{1}{2}$	$5\frac{1}{2}$	7	7	$8\frac{1}{2}$	$8\frac{1}{2}$	10	10	$11\frac{3}{4}$	$11\frac{3}{4}$
F	3	$3\frac{5}{8}$	3	$3\frac{5}{8}$	3	$3\frac{5}{8}$	3	$3\frac{5}{8}$	3	$3\frac{1}{2}$
G	3	$2\frac{3}{8}$	3	$2\frac{3}{8}$	3	$2\frac{3}{8}$	3	$2\frac{3}{8}$	3	$2\frac{1}{2}$
H	6	6	6	6	6	6	6	6	6	6
J	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$ •	$2\frac{5}{16}$
Size of Pipe Taps	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	2	2
No. of Tubes	8	15	10	19	12	23	14	27	16	31

Table 106. *Air Friction through Trane Coils with ½-Inch Tubes Having a Dry Surface

Air friction is given in inches of water. Air velocities are based on air having a dry-bulb temperature of 70°F. and 50% relative humidity.

No. of Rows of Tubes	Face Velocity in Feet per Minute							
	200	300	400	500	600	700	800	1000
1	.012	.025	.043	.070	.095	.13	.165	.26
2	.02	.040	.068	.105	.145	.19	.24	.36
3	.026	.055	.090	.143	.195	.253	.315	.47
4	.033	.070	.115	.180	.242	.313	.390	.573
6	.045	.095	.160	.253	.340	.437	.54	.785
8	.057	.120	.205	.327	.440	.56	.69	.995

*Courtesy of The Trane Company, LaCrosse, Wisconsin.

Table 107. *Increased Air Friction Due to Wet Surface

Initial Air Dew Point Minus Initial Water Temperature	Factor for Friction Increase	Initial Air Dew Point Minus Initial Water Temperature	Factor for Friction Increase
10	1.1	40	1.37
20	1.19	50	1.40
30	1.28	..	....

Table 108. *Table for Finding the Resulting Dry- and Wet-Bulb Temperature of a Mixture of Return Air and Outside Air Entering the Cooling Coils

To be used only for return air having an 80° dry-bulb temperature and 50% relative humidity.

Percentage of Outside Air	Dry-Bulb Temperature of the Air Mixture			Wet-Bulb Temperature of the Air Mixture						
	Outside Dry-Bulb Temp.			Outside Wet-Bulb Temperature						
	95	100	105	74	75	76	77	78	79	80
Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9	Col. 10	Col. 11
2	80.3	80.4	80.5	67.2	67.2	67.2	67.2	67.3	67.3	67.3
4	80.6	80.8	81.0	67.3	67.3	67.4	67.4	67.5	67.5	67.6
6	80.9	81.2	81.5	67.5	67.5	67.6	67.7	67.7	67.8	67.9
8	81.2	81.6	82.0	67.6	67.7	67.8	67.9	68.0	68.1	68.2
10	81.5	82.0	82.5	67.8	67.9	68.0	68.1	68.2	68.4	68.5
12	81.8	82.4	83.0	67.9	68.0	68.2	68.3	68.5	68.6	68.8
14	82.1	82.8	83.5	68.1	68.2	68.4	68.5	68.7	68.9	69.1
16	82.4	83.2	84.0	68.2	68.4	68.6	68.8	69.0	69.2	69.4
18	82.7	83.6	84.5	68.3	68.5	68.8	69.0	69.2	69.4	69.6
20	83.0	84.0	85.0	68.5	68.7	69.0	69.2	69.4	69.7	69.9
22	83.3	84.4	85.5	68.6	68.9	69.1	69.4	69.7	69.9	70.2
24	83.6	84.8	86.0	68.8	69.1	69.3	69.6	69.9	70.2	70.5
26	83.9	85.2	86.5	68.9	69.2	69.5	69.8	70.1	70.4	70.8
28	84.2	85.6	87.0	69.1	69.4	69.7	70.0	70.4	70.7	71.0
30	84.5	86.0	87.5	69.2	69.6	69.9	70.2	70.6	70.9	71.3

*Courtesy of The Trane Company, La Crosse, Wis.

Table 109. *Required Wet=Bulb Temperature of the Air Supplied to a Conditioned Space for Various Percentages of Sensible Heat Gain

To be used only for room conditions of 80°F. dry-bulb temperature and 50% relative humidity.

D.B. Temp. of Air Supply	C.f.m. per 1000 B.t.u. Sensible Heat Gain per Hr.	Face Area of Coil Required for Each 1000 B.t.u. of Sensible Heat Gain per Hour — Face Velocity Ft. per Min.			Wet-Bulb Temperature of the Air Supply — Sensible Heat Percentage									
		400	500	600	100%	95%	90%	85%	80%	75%	70%	65%	60%	55%
Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9	Col. 10	Col. 11	Col. 12	Col. 13	Col. 14	Col. 15
40	22.62	.0566	.0452	.0377	......	......	......	......	......	......	......	......	39.0	35.7
41	23.20	.0580	.0464	.0387	......	......	......	......	......	......	......	......	40.0	36.7
42	23.81	.0595	.0476	.0397	......	......	......	......	......	......	......	......	40.9	37.7
43	24.45	.0611	.0489	.0408	......	......	......	......	......	......	......	......	41.7	38.8
44	25.13	.0628	.0503	.0419	......	......	......	......	......	......	......	......	42.6	39.7
45	25.85	.0646	.0517	.0431	......	......	......	......	......	......	......	......	43.4	40.7
46	26.61	.0665	.0532	.0444	......	......	......	......	......	......	......	......	44.3	41.7
47	27.42	.0686	.0548	.0457	......	......	......	......	......	......	......	47.7	45.1	42.6
48	28.27	.0707	.0565	.0471	......	......	......	......	......	......	......	47.9	45.9	43.5
49	29.19	.0730	.0584	.0487	......	......	......	......	......	......	......	48.6	46.7	44.4
50	30.16	.0754	.0603	.0503	......	......	......	......	......	......	......	49.3	47.5	45.4
51	31.20	.0780	.0624	.0520	......	......	......	......	......	......	......	50.0	48.3	46.3
52	32.31	.0808	.0646	.0539	......	......	......	......	......	......	52.2	50.7	49.1	47.1
53	33.51	.0838	.0670	.0559	......	......	......	......	......	......	52.7	51.4	49.8	48.0
54	34.80	.0870	.0696	.0580	......	......	......	......	......	54.7	53.3	52.1	50.6	48.8
55	36.19	.0905	.0724	.0603	......	......	......	......	......	54.9	53.9	52.7	51.3	49.6
56	37.70	.0943	.0754	.0628	......	......	......	......	56.5	55.4	54.5	53.4	52.0	50.5
57	39.34	.0984	.0787	.0656	......	......	......	57.8	56.8	56.0	55.1	54.0	52.8	51.3
58	41.13	.1029	.0823	.0686	......	......	58.7	57.9	57.3	56.5	55.7	54.7	53.5	52.1
59	43.08	.1077	.0862	.0718	......	59.5	58.9	58.3	57.8	57.1	56.3	55.3	54.2	52.8
60	45.24	.1131	.0905	.0754	60.2	59.7	59.3	58.8	58.2	57.6	56.8	55.9	54.9	53.6
61	47.62	.1191	.0952	.0794	60.5	60.1	59.7	59.2	58.7	58.1	57.4	56.6	55.6	54.4
62	50.26	.1257	.1005	.0838	60.9	60.5	60.1	59.7	59.2	58.6	58.0	57.2	56.2	55.1
63	53.22	.1331	.1064	.0887	61.2	60.9	60.5	60.1	59.7	59.1	58.5	57.8	56.9	55.9
64	56.55	.1414	.1131	.0943	61.6	61.3	60.9	60.5	60.1	59.6	59.0	58.4	57.6	56.6
65	60.32	.1508	.1206	.1005	61.9	61.6	61.3	61.0	60.6	60.1	59.6	58.9	58.2	57.3
66	64.63	.1616	.1293	.1077	62.3	62.0	61.7	61.4	61.0	60.6	60.1	59.5	58.8	58.0
67	69.60	.1740	.1392	.1160	62.6	62.4	62.1	61.8	61.5	61.1	60.6	60.1	59.5	58.7
68	75.40	.1885	.1508	.1257	63.0	62.8	62.5	62.2	61.9	61.6	61.2	60.7	60.1	59.4
69	82.25	.2056	.1645	.1371	63.3	63.1	62.9	62.7	62.4	62.0	61.7	61.2	60.7	60.1

The air is saturated at the temperature shown by the bold face figures.
The bold face figures in the above table represent both the dry- and wet-bulb temperatures of the air supply since this air is saturated. Do *not* use bold face figures in connection with dry-bulb temperatures given in Column 1.

Table 110. *Air Friction through Trane Coils with ¾-Inch Tubes Having a Dry Surface

Air friction is given in inches of water. Air velocities are based on air having a dry-bulb temperature of 70°F. and 50% relative humidity.

No. of Rows of Tubes	Face Velocity in Feet per Minute							
	200	300	400	500	600	700	800	1000
1	.018	.032	.055	.080	.110	.145	.185	.27
2	.025	.048	.080	.120	.155	.210	.26	.39
3	.032	.060	.100	.153	.210	.270	.335	.51
4	.040	.075	.125	.190	.257	.332	.41	.63
6	.057	.100	.173	.263	.352	.46	.56	.87
8	.072	.13	.220	.338	.450	.587	.71	1.11

*Courtesy of The Trane Company, La Crosse, Wis.

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